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CONTROLLED DIRECTIONAL SOLIDIFICATION.

By GEORGE BATTY.*

The term "controlled directional solidification" was used by Briggs and Gezelius in their paper † to define a principle which they believe is of major importance in the production of steel castings. It is the opinion of the writer that this principle may be found valuable in the production of cast pieces produced in metals other than steel.

The term used by Briggs and Gezelius is much more definitive and comprehensive than is the more customary foundry term "progressive solidification" and it prescribes procedure that is devised to promote integral soundness of cast structures.

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† "Studies on Solidification and Contraction and Their Relation to the Formation of Hot Tears in Steel Castings." By Charles W. Briggs and Roy A. Gezelius, Division of Physical Metallurgy, U. S. Naval Research Laboratory, Bellevue, Anacostia, D. C., presented at the A. F. A. Convention, Chicago, June, 1933.

In the following pages it is hoped to place information before engineers and designers which may influence them in their designs to produce or prescribe structures which can be made integrally sound or free from internal hot tears or cavities.

Before proceeding with the argument the writer wishes to state that the original conception which has led to the harnessing of certain fundamental facts was not his own, and the reader's indulgence is craved for a citation of the matters which led the author to investigate the potentialities of promoting a definite and controllable temperature gradient in both mold and metal with a view to producing integrally sound steel castings of various well-defined types.

In 1921 the writer was attached to the foundry in the South of England which was under the management of Mr. F. A. Melmoth and in that foundry he observed, on one particular type of casting, the application of the principle of delivering metal to a feed head located immediately above the ingate. This rigging was devised by an excellent practical foundryman, Mr. Harry Langley, and was the result of repeated experiments in the production of half-gear-blank castings.

This type of casting is illustrated in Figure 1 and it is interesting to note that external chilling was also applied in an endeavor to promote the desired and requisite temperature gradient in the metal.

It should be stated that the scheme indicated in Figure 1 was not entirely satisfactory in respect to the chilling of the heavy sections on the flat side of the half-gear-blank and the writer later modified this chilling system to achieve the desired results.

Another important feature was not used, at that time, but was introduced by Melmoth early in 1922. This consisted of tilting the mold to an angle of about 15 degrees for pouring as indicated on Figure 2, the object being to cool the metal progressively in its transit through the mold so that the effect of the external chill at the hub would be more efficient. After pouring was completed the mold was lowered off to the horizontal position. It was found that this procedure amplified the efficiency of the gating and heading system and produced sounder castings.

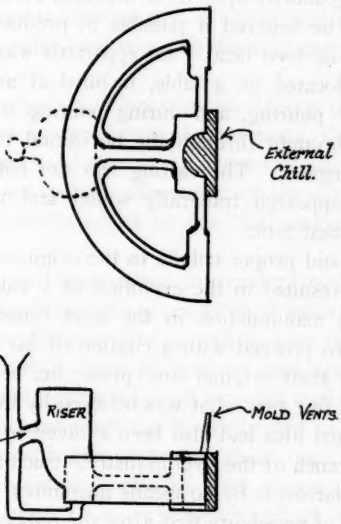

FIG. 1.

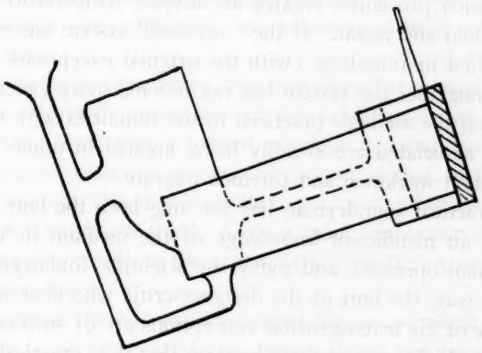

FIG. 2.

Following shortly upon these findings, Melmoth remarked to the writer that he believed it possible to produce a full ring casting with only one feed head. An apparatus was rigged whereby the mold was located on a table, inclined at an angle of about 15 degrees for pouring, and during pouring the table carrying the mold was brought through the horizontal to a reverse angle of about 15 degrees. The casting was not entirely satisfactory although it appeared integrally sound and the method was not pursued at that time.

Having paid proper tribute to the originators of the conception which has resulted in the evolution of a valuable system of procedure and manipulation in the steel foundry, the writer feels competent to proceed with a citation of his own development, in practice, of their original conception; but reference will be made later to the fact that what was believed by the writer to be an entirely original idea had also been evolved by another worker in a different branch of the steel industry. Such an example of unconscious plagiarism is by no means uncommon and it is even within the bounds of possibility that after the publication of these notes a multitude of witnesses will arise to proclaim that there is nothing new in the practices expounded. The system of "banking" molds for pouring is quite common and, gated at the low end of the casting, such procedure creates an adverse temperature gradient in both mold and metal. If the "reversal" system has previously been applied in foundries (with the original exceptions noted) it is surprising that the system has not become more general. It is known that the methods practiced in the foundries with which the writer is associated are steadily being applied in other foundries as competent workmen and foremen migrate.

The practical foundryman has for long been the butt of critics who had an insufficient knowledge of the medium in which the foundryman operated, and today the scientific foundryman is, in a similar way, the butt of the designer-critic who does not realize that some of his heterogeneous conceptions are of such complexity of section thickness and mass location that it is practically impossible to produce a sound casting of the predicated design. Happily we now see—probably as a result of the desire of designers to use built-up structures in the form of weldings—somewhat a changed

attitude in the designer's mind, and, as foundrymen, are convinced that cooperation between founder and designer will promote the production of better, safer, and more serviceable, cast components in all types of metals.

In their work on ingots and ingot molds Harry Brearley and Arthur Brearley used a very simple diagrammatic illustration to indicate the solidification of steel in an ingot mold and used this illustration to demonstrate the advantage of the "big end up" mold as against the parallel-sided or the "bid end down" ingot mold.

It is illuminating and instructive to ponder the tremendous amount of thought which has been given to ingot mold design since the publication of the Brearleys' work and to note the insistence upon the inter-relation of ingot dimensions and upon the controlled graduated chilling effect achieved by ingot molds of specific design. This thought and work has been directed to the production of integrally sound ingots and when these ingot designs are compared with many simple forms of steel castings it becomes apparent that the designer, by inference, endows the castings founder with an intelligence and a technique immeasurably superior to that endowed, by inference, upon the ingot founder. There is no such difference, and steel founders are regularly placed in the position, in discussion with designers, of urging certain facts which unfortunately are regarded as excuses instead of definite explanations.

There is no controverting the fact that for certain practical purposes Brearley's conception of the solidification of steel as a continuously thickening solidified envelope is satisfactory; nor is there any good reason to dispute the contention that, in the making of steel castings, the metal solidifies as a result of transference of heat from itself to the mold. Only a small proportion of the heat is transferred directly to the atmosphere from the exposed surface of heads or risers.

The practical foundryman has noted that metal which is flowing up in heads or risers relatively remote from the ingate is very much cooler than the metal which is flowing into the mold from the ladle. It is obvious, therefore, that this metal in the remote risers has been appreciably cooled in its transit through the mold

and it naturally follows that such cooling of metal is indicative of an equivalent heating of the mold. Such heating of the mold must be, to some extent, proportional to the duration of contact of metal with mold and, hence, it is easy to visualize the temperature gradient created in the mold in the actual process of filling the mold with metal.

Because of the nature of the molding media available it is fairly general practice to lead the metal, by ingate or ingates, into the lower levels of the mold and it follows necessarily that, when the mold is filled, there has been created an adverse temperature gradient in both mold and metal, the lower levels of the mold being hot and containing hot metal whilst the upper levels of the mold are relatively cool and contain relatively cool metal. By the use of risers or feed heads of a section appreciably greater than the sections upon which they are imposed, and by reason of the practice of filling up feed heads from the top, direct from the pouring ladle, an appearance of satisfactory soundness is secured when the feed heads are removed. It is often found, however, that the lower levels (as cast) of steel castings are afflicted with internal shrinkage cavities or internal hot tears.

The reason for this condition may be illustrated by the application of Brearley's diagramatic illustration of the freezing of steel ingots but, instead of ingot molds, simple sections in a sand mold are postulated. Figures 3, 4, and 5, indicate three identical sections or castings of an ingot-like form and differ only in the system of heading and gating. The section illustrated is typical of the upper part, or half, of many steel castings produced from split patterns which must necessarily be " drafted" to permit withdrawal from the sand. They are, in effect, of the " big end down" ingot type and therefore tend to promote the internal or axial looseness which the " big end up" ingot mold was devised to eliminate.

In Figure 3 the example is top poured, there being no sand ingate to the mold. In Figure 4 the example is bottom gated as is customary with most steel castings, whilst in Figure 5 the example is bottom gated into the riser or feed head and, after pouring is completed, the mold is reversed through an angle of 180 degrees to bring the riser into its proper feeding position in relation to the casting.

It should here be stated that the writer had been operating the "total reversal system" on steel castings of certain types for many months before he saw specifications of a patent taken out by H. Brearley for the production of steel ingots, such patent having reference to the bottom-pouring of a row of ingots in a "cradle," the lower part of each ingot mold being lined with refractory material (as the orthodox hot-top), and the whole set of molds, in the "cradle," reversed through an angle of 180 degrees as soon as the vents of the molds and the mouth of the "trumpet" were sealed.

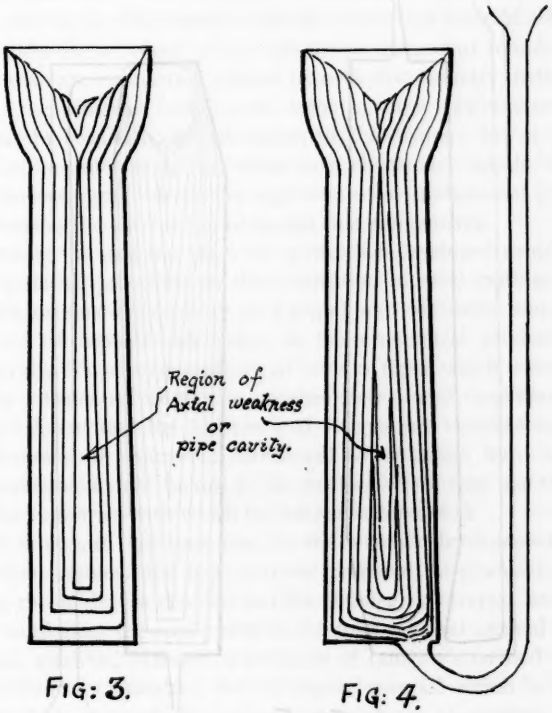

FIG: 3. FIG: 4.

The progress of solidification is indicated on Figures 3, 4, and 5 in relation to the comparative degree of the heating of the mold by the metal in the process of filling the mold and it will be seen that while example, Figure 3, presents evidence of little, or no adverse temperature gradient, example Figure 4 shows considerable evidence of such an adverse temperature gradient in both mold and metal. It is not affirmed that the solidification lines present an absolutely accurate picture but they are certainly roughly indicative of what actually occurs.

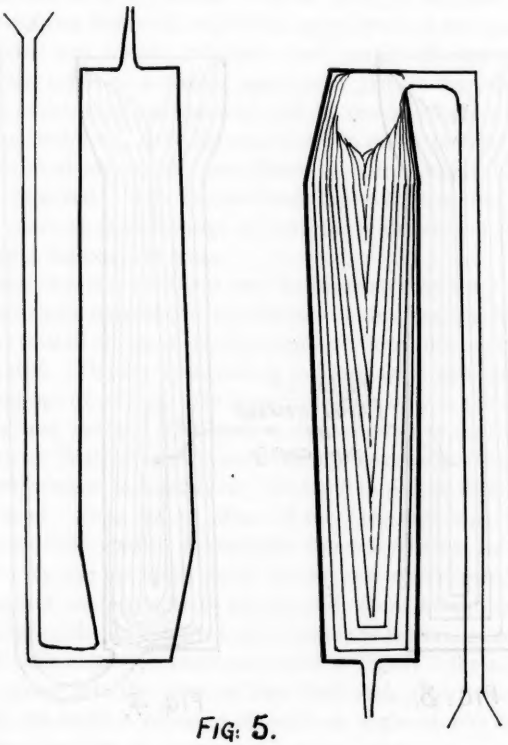

FIG. 5.

The example, Figure 5, indicates that by the use of the reversal system of mold manipulation advantages are secured which are greater than those accruing from simple open top "big end up" pouring and, at the same time, full advantage is taken of the system of bottom gating which permits the quiet introduction of metal to the mold and eliminates dangers from cutting and splashing. In Figure 5 we have an illustration of what may be considered the most desirable design and procedure in the production of steel castings but it is readily conceded that such simplicity of design is very rarely available to the steel founder. Yet the fact remains that a good many steel castings are of such design that this principle may be applied cheaply with the result that considerable cash economies are achieved, along with the production of sounder castings, than are regularly produced by orthodox foundry methods.

To the practical foundryman some of these explanations may appear to be a tedious elaboration of the obvious, but it is considered desirable to go into some most elementary details in order to convince certain designing engineers of a fundamental principle underlying the production of sound cast components.

During the past ten years the writer has developed simple mold manipulative procedure in the production of steel castings, based on the original conceptions of Langley and Melmoth, which have proved of considerable value in the economical production of integrally sound steel castings of certain types which were previously difficult to produce in an absolutely sound condition. It is also believed that the insistence of Briggs and Gezelius upon the importance of controlled directional solidification may, to some considerable extent, be due to the practice which they had observed in the foundries with which the author is connected.

It is almost ludicrous that, in the writer's development of the reversal systems, the total reversal preceded, in practical application, the 30 degree reversal and the 100 degree reversal, and it was not until after the application of the total reversal method that the writer modified Melmoth's variation of Langley's method on half-gear-blanks to produce the 30 degree reversal which is simplest and which can easily be applied to many types of castings.

In 1924 while the writer was Works Manager and Technical Director of the National Steel Foundry (1914) Ltd., of Leven,

Scotland, a front wheel for Thorneycroft trucks was being made and ordinary methods of foundry procedure did not produce castings satisfactorily sound at the hub plate.

This wheel was of the cruciform section spoke type and is illustrated, in section, in Figure 6. It should be remembered that in 1924 we did not have the knowledge of synthetic sands which we now possess and that certain natural-bonded sands were not available at a proper price. The wheel, therefore, had to be poured in the position indicated on Figure 6 in order that metal could be quietly introduced at the lower levels of the mold.

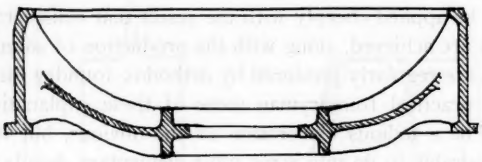

FIG: 6

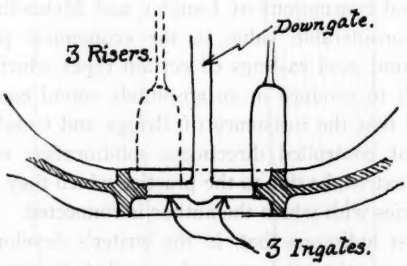

FIG: 7

Pouring the casting in the reversed position, so that the metal flowed downhill from the hub plate, was not practicable because of the nature of the molding sand. Further, it was necessary to gate these wheels uniformly from the center in order to minimize any tendency to distortion, the wheels having to be supplied in the fully machined condition and balanced to close weight limitations.

Green sand practice was essential because it was a matter of experience that these relatively light-section castings became overstressed as a result of mold resistance to normal contraction if dry sand practice was followed.

Orthodox procedure in heading and gating is indicated on Figure 7, there being three feed heads imposed immediately above the ingates. This of itself was sound procedure but, because of the design of the casting, the system was found to be inadequate. Beneath each riser, imposed on the hub plate, was a small contraction cavity and it was found that increasing both the height and the volume of these risers did not cure the defect. It was considered inexpedient to proceed on lines of production which would necessitate repairing, by welding, each casting at three places on the hub plate.

Sectioned castings cut into as many as forty pieces showed the structure to be satisfactorily sound at all places except the three spots under the risers immediately opposite the ingates and this of itself is evidence of the beneficial effect of progressive cooling of the metal—in relation to the production of a sound structure—in its passage through the mold. The rim of the wheel was of a section approximately equivalent to the spoke section and the rim, on cutting up, appeared perfectly sound.

Visualizing the actual filling of the mold, as illustrated on Figure 7, it is seen that the metal in the feed heads is progressively cooled as it rises in the mold whilst a definite hot spot is created at each ingate. On sectioning a riser it became apparent that risers so positioned could not function efficiently because of the hot spot and, if no risers were imposed immediately above the ingates, the condition would be even worse. A complete annular riser was tried instead of the three separate feed heads but this procedure proved no more efficient and the unorthodox system indicated on Figure 8 was adopted with perfect success.

An annular riser was placed below the hub plate and into this riser the ingates were led as shown on Figure 8. The casting was poured and the ladle moved away; after an interval of 40 seconds had elapsed the downgate was sealed with wet sand, the mold picked up by a crane and reversed through an angle of 180 degrees. So efficient was this system of pouring and manipulation that the

dimensions of the feed head were reduced so that the head and gate discard amounted to only 13 per cent of the total metal put into the mold, whereas the previous inefficient method had involved a discard of head and gate metal totaling 26 per cent.

This casting provides an example of controlled directional solidification and led to the adoption of mold manipulative procedure on many other types of castings.

This particular casting suffered, in the original, orthodox method from defects which arose as a result of hot spots closely adjacent to the ingates and a scrutiny of Figure 7 will indicate that, because of the design of the casting, the metal which filled the risers was rising only slowly in these cavities and was being progressively subjected to the cooling action of the mold in precisely the same manner as was the metal which produced the spokes and rim of the wheel.

Within each riser or feed head there was produced a distinctly adverse temperature gradient similar to that indicated on Figure 4. The intensity of the hot spot adjacent to each ingate was such that even the large feed heads could not produce a sound condition.

By comparison, the method indicated on Figure 8 locates the hot spots entirely outside the casting proper, and it can readily be

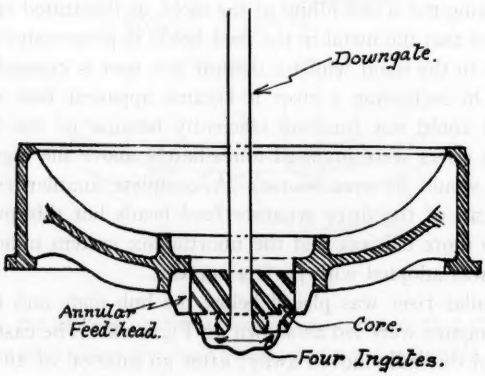

FIG. 8.

seen that in the process of filling the mold the feed head cavity is thoroughly heated by having passed through it the whole of the metal which produces the casting. After the mold is reversed there is produced, within the feed head and the central or hub portion of the casting, a pronouncedly favorable temperature gradient in both mold and metal which predicates, throughout the whole casting, controlled directional solidification. In visualizing the filling of a mold with metal we may, for our present purpose, plot the rise of metal in mold as a series of horizontal lines although it is well understood by most practical foundrymen that in large castings the metal, rising in the mold, is actually flowing in a series of waves under the impulse of the superimposed metal in the downgate.

One more example of a "total reversal" will be given before proceeding to detail the simpler procedure for the systems which have been called "30-degree reversal" and "100-degree reversal."

The first application which the writer made in this country of what is now called controlled directional solidification was upon a brake drum casting which had given considerable trouble in manufacture.

Whereas the truck wheel previously described is an example of relatively harmonious design and fairly uniform section, the brake drum now to be discussed provides an example of differential sections such as normally may lead to troubles from internal cavities when produced on orthodox lines.

The orthodox procedure of gating and heading is indicated on Figure 9 and it may here be stated even with the extravagant heading of these castings, a disastrous proportion of the castings was found to be unsound upon machining. This of course was due to the hot spots in the region of the two ingates and the internal machining of the braking face most frequently revealed shrinkage cavities at these locations. Only occasionally did the external machining of the "rim" of the drum reveal these cavities. In addition it was frequently found that the feed heads on the bolting face or hub of the drum had not functioned efficiently and that small cavities had to be repaired by welding.

The "lumped" risers imposed on the drum in an attempt to overcome the ingate hot spots never functioned efficiently.

Figure 10 indicates the arrangement for total reversal and the casting is shown in the pouring position. The feed head was calculated to have a volume equal to 13 per cent of the volume of the casting and the downgate was 1¼-inch diameter in order to prevent the casting being poured too rapidly.

An essential feature of the reversal system of pouring is that the molds shall be filled as slowly as is practicable, using metal as hot as is practicable. This is prescribed in order that the greatest practicable temperature gradient may be promoted in both mold and metal because the system provides the means whereby such a temperature gradient may be used to its greatest potential advantage.

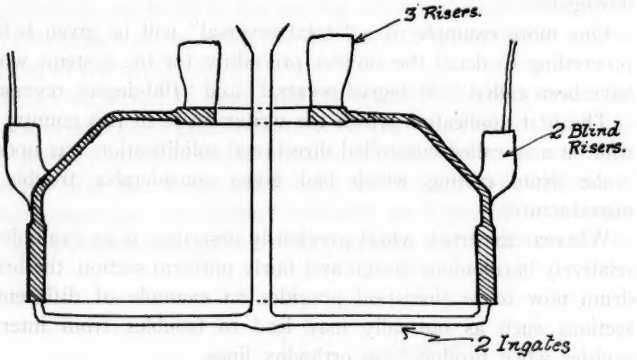

FIG. 9.

In producing the brake drum, as illustrated on Figure 10, it will be noted that the actual ingate to the casting is the junction of the head with the casting and this "gate aperture" is so very much larger than the orthodox ingates that the tendency to serious mold erosion at the ingate is almost entirely eliminated.

The pouring procedure, in the foundry in which these castings were made, was quite simple as a small crane was available for reversing the molds after pouring. On a casting of this type the reversal must be accomplished promptly and all that is necessary

is to seal the head of the downgate with a handful of wet sand and turn the mold over on the previously provided trunnions. The mold is turned through an angle of 180 degrees in the direction indicated on Figure 10, such direction being mandatory because of the danger of bleeding the head if it were reversed in the opposite direction and the seal failed to hold. After reversing through 180 degrees the feed head is in a superior position and functions efficiently.

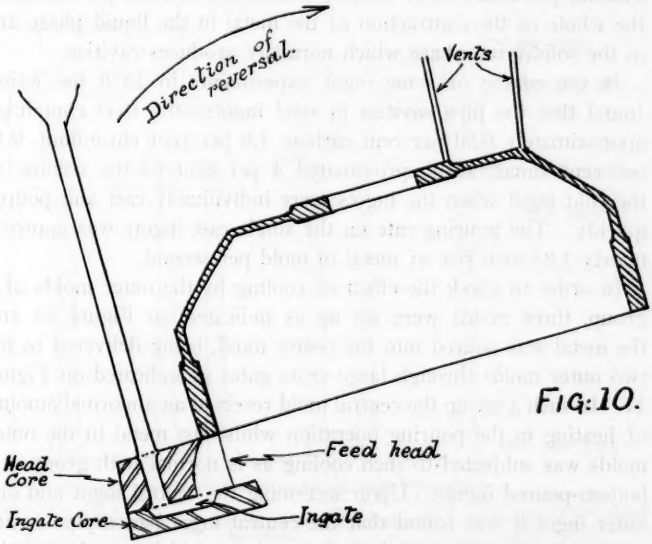

FIG. 10.

It was found in actual practice that the temperature differential in both mold and metal was adequate to produce a satisfactory degree of integral soundness in the relatively heavy hub section or braking face of the casting but it is believed that such a satisfactory state of affairs would not uniformly pertain if the molds were filled rapidly, because such rapid filling of the mold would reduce the temperature gradient existent in both mold and metal.

In the economical production of steel castings the amount of discard in the form of heads and gates must be considered as a potent factor. This discard is available for remelting but as such

has a value approximately equal to that of the regular scrap materials purchased in the open market. It must be remembered, however, that liquid metal in the ladle has a value considerably greater than has unmelted scrap metal and it therefore becomes apparent that all such liquid metal converted into gates and risers suffers a serious depreciation in value. In electric furnace practice this depreciation in value may be as high as 1½ cents per pound. It therefore becomes necessary so to organize molding, gating, and heading procedure as to contain within the smallest possible head the whole of the contraction of the metal in the liquid phase and in the solidifying phase which normally produces cavities.

In the course of some ingot experiments in 1920 the writer found that the pipe cavities in steel ingots—the steel containing approximately 0.30 per cent carbon, 1.0 per cent chromium, 0.15 per cent vanadium—approximated 4 per cent of the volume of the cold ingot when the ingots were individually cast and poured quickly. The pouring rate on the single-cast ingots was approximately 1.25-inch rise of metal in mold per second.

In order to check the effect of cooling by the outer molds of a group, three molds were set up as indicated on Figure 11 and the metal was poured into the center mold, being delivered to the two outer molds through large cross gates as indicated on Figure 11. In such a set-up the central mold received an abnormal amount of heating in the pouring operation whilst the metal in the outer molds was subjected to such cooling as is normal with group-cast bottom-poured ingots. Upon sectioning the central ingot and one outer ingot it was found that the central ingot had a pipe cavity equal to 3.25 per cent of the volume of the cold ingot whereas the outer ingots had a pipe cavity equal to only 1.7 per cent of the volume of the cold ingot.

The observed pouring temperature at the nozzle was 2700 degrees F. and the observed temperature on the surface of the metal in an outer mold when the mold was about half filled, was 2550 degrees F. It is not affirmed that these temperatures, being taken with an optical pyrometer, were correct but it is probable that the difference in the two temperatures was approximately correct.

In the conditions which pertain in bottom-poured ingots it is not safe to assume that a temperature reading on the surface metal

in the mold will be an accurate index of the whole of the metal in the mold at the time the observation was made, nor will be registered precisely the average temperature of the metal if the mean of temperatures at the nozzle and at the surface of an outer ingot be taken. The observed temperature on the surface in an outer mold may only be considered as representing the lowest temperature pertaining in the fluid steel during the pouring operation; the average temperature of the metal in the outer mold will

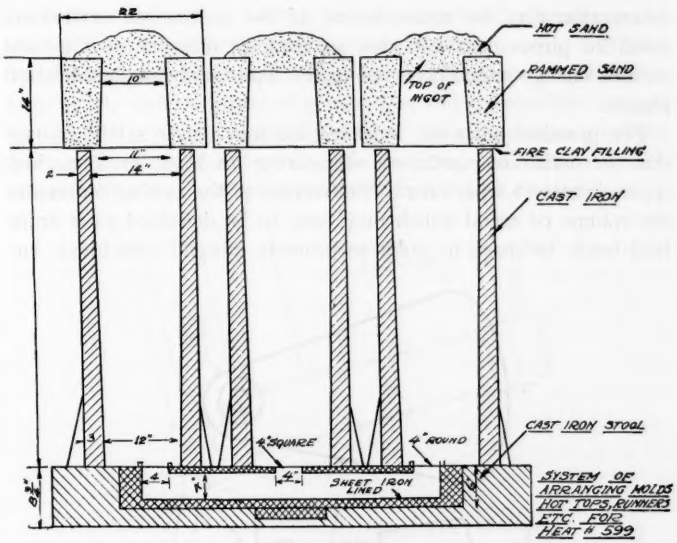

FIG: 11.

be somewhere—indeterminable—between that temeprature and the heat of the metal as it leaves the ladle. The observed difference of temperature between the stream from the ladle and the surface metal in an outer mold was 150 degrees F. and this must be related to the decreased amount of contraction cavity shown in the outer ingot as compared with the central ingot.

In pouring this group of ingots the pouring speed was approximately .5 inch per second, only approximately 40 per cent of the

pouring speed of the single cast ingots which showed contraction cavities of approximately 4 per cent.

It is not suggested that these figures concerning contraction cavities properly express the total fluid and solidification contraction of steel because it is realized that as soon as a solidified envelope is produced sufficiently thick and strong to withstand the ferrostatic pressure of the liquid interior, the ingot or casting commences to constrict and we may therefore visualize a slightly constricting vessel containing the cooling liquid, which in turn will lead to the assumption that the measurement of the contraction cavities of solidified pieces does not give an accurate measure of the total volume change which occurred in the liquid and the solidification phases.

For practical purposes, however, we may fairly safely assume that, in orthodox conditions of pouring an ingot or a casting, approximately 4.5 per cent of the volume of the casting represents the volume of metal which may have to be delivered to it from feed heads or risers in order to promote integral soundness; but

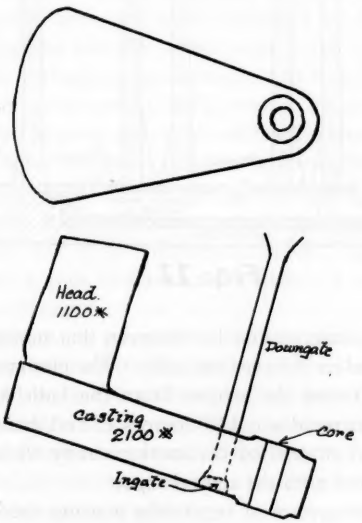

FIG. 12.

the ingot experiments just cited indicate that on certain types of castings a considerable amount of the fluid contraction may safely be accomplished in the actual process of pouring provided that an adverse temperature gradient in both mold and metal be avoided.

The total reversal system of pouring which has already been illustrated provides what is probably the ideal method of making many types of steel castings because that part of the mold which surrounds the feed head is actually converted into a hot top in the process of pouring and a definite and pronounced temperature gradient is produced in both mold and metal which operates in the correct direction when the mold is reversed after pouring. It must be admitted, however, that manipulative difficulties are bound to arise as the size of the casting increases and it therefore follows that in order to achieve the benefits of the temperature gradient that permits or produces controlled directional solidification, much simpler and easier manipulative methods must be devised.

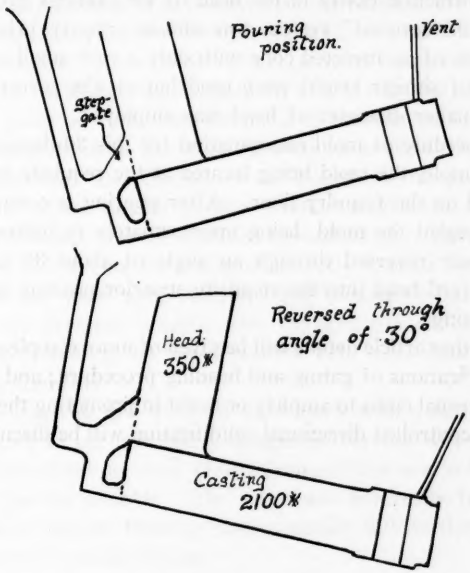

FIG: 13

An early example of the 30-degree reversal is the block shown on Figure 12, the lower section indicating the orthodox gating and heading procedure with the casting poured in the orthodox "banked" position. In orthodox procedure the feed head weighed approximately 1100 pounds and the casting 2100 pounds.

By adopting the 30-degree reversal system as indicated on Figure 13, and promoting thereby controlled directional solidification, it was possible to reduce the head discard from 1100 pounds to 350 pounds and, consonantly to reduce cleaning room expenses in respect of cutting and grinding.

Heads were sectioned from castings made on the orthodox method and on the new method, and a striking difference in the shape of the contraction cavity contained in the head was apparent. The head from the casting produced in the orthodox manner had a relatively shallow primary contraction cavity and a long narrow secondary pipe similar to those observable in parallel ingots.

The contraction cavity in the head of the castings produced on the "partial reversal" system was almost entirely primary pipe in the form of an inverted cone with only a very small secondary.

Heads of similar height were used but on the reversal system a much smaller diameter of head was employed.

The procedure of mold manipulation for this 30-degree reversal is quite simple, the mold being located at the requisite angle on a bar or rail on the foundry floor. After pouring is completed and the vent sealed the mold, being approximately in balance on the bar, is easily reversed through an angle of about 30 degrees to bring the feed head into the requisite superior position in relation to the casting.

In a further article details will be given of more complex castings, with modifications of gating and heading procedure; and the function of external chills to amplify or assist in promoting the requisite degree of controlled directional solidification will be discussed.

THE FUNDAMENTALS OF ELECTRIC ARC WELDING.

By Bela Ronay, Civil Member.*

Electric arc welding, a means of fabricating practically all component parts of Naval vessels, is a procedure that is taken for granted. Many — even technical men — who use welding as a method of construction have but a hazy idea of how the metal of the electrode—the welding rod—becomes transferred into weld metal of the deposit.

For a long time there were many, and there are still a few diehards, who believe that the metal is transferred in the form of vapor, simply because it is known that the temperature of the arc itself is somewhere above 5000 degrees F., at which temperature it may volatilize. As a matter of fact, only an infinitesimally small part of the metal becomes volatilized and it may be said, from a practical point of view, that the metal is transferred from the tip of the electrode upon the work as a liquid in the form of drops. The temperature and fluidity of the liquid are factors of prime importance in the quality of the resultant weld metal.

Let us consider first the so-called bare electrode. Bare electrodes were used in this country almost exclusively in most welding work till about five years ago, when welding became of interest to the builders of pressure vessels, who changed the entire aspect of electric arc welding by developments that were necessitated by their requirements.

Bare electrodes are wires of mild steel, with a carbon content of 0.04 to 0.18 per cent, with manganese varying correspondingly from 0.12 to 0.60 per cent, phosphorous, silicon and sulphur being kept as low as possible. The wires are not really bare, as the lubricant of the last drawing pass is usually left on them, forming a thin layer of oxide coating.

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When welding is performed with a bare electrode, the following phenomena take place: The heat of the arc heats the tip of the electrode. With increasing temperature the metal begins to liquefy and the liquefied metal begins to form a drop. This drop formation takes place in exactly the same manner as it does at the end of a melting icicle. When a drop of metal reaches a definite volume, then gravity and the magnetic forces present tear it away from the tip of the electrode and deposit it on the work. This cycle repeats itself with a fairly constant frequency of an order of from 15 to 30 per second, depending on the type and size electrode used. There are several ways that enable the inquirer to verify this statement. Oscillograms of electric arc welds give the results of these drop sequences by recording the potential changes during the state of existence of each drop. High potential values exist during the formation periods of each drop—when there is an almost purely electric arc condition. Low potential periods refer to the transfer time, when a metal-to-metal bridge through the arc practically short-circuits it.

Another, and more direct, way of verifying the statement is by high speed photography. High speed films projected at normal speed make the observation of the birth, growth, and finally the transfer, of each drop convenient and positively convincing.

Another, and less expensive, but equally instructive method is the use of a stroboscope when trained directly upon the arc. This method is especially valuable when a high grade electrode is used in a sensitive automatic welding machine, as in such cases the stroboscope is easily adjusted and instead of brief snatches, the drop development may be observed for a considerable length of time at any phase of the formation of the drops. This method is perhaps the most convincing and the least expensive, but unfortunately leaves no record.

The performance of the electrode, that is the quality of the resultant metal, as well as the nature of the fusion between bare metal and deposited metal, depends a great deal on the volume of the individual drop, the drop frequency, and the time relationship between the formation period and transfer period.

How does the volume of the individual drop affect the quality of the deposited metal or the bond of fusion? Each drop of metal

that leaves the tip of the electrode has to come to rest in a pool of molten metal in order that the base metal and deposited metal shall fuse as an alloy. Regardless of the fact that the drop of metal that leaves the tip of the electrode is perhaps superheated, it would not fuse with the base metal if this were cold enough to chill it. The electric energy input can not be more than necessary to melt the electrode at such a rate as may be conveniently handled by the operator. The same heat that is liberated by the arc has to melt the base metal also. This, however, is of a volume many times greater than the affected part of the electrode and the heat conduction in the base metal is greater than it is in the unidirectional electrode. Therefore, it may happen that the volume of the molten pool of base metal is less than that of the corresponding individual drop of electrode metal.

In addition, it may be expected that the molten metal of the base metal, by the same reason, is of lower temperature. The result, in such a case, would be a pasty conglomerate and if such a condition were repeated throughout the sequence of the drops, the resulting weld would be of poor fusion—alas, too well known to many of us. Consequently, the electrode should melt in drops of minimum volume and weight, to insure good fusion.

The volume of the individual drop of a bare electrode depends upon the following factors: (*a*) Heat input. (*b*) Chemical composition. These factors, in turn, are dependent upon the fluidity and surface tension of the molten metal. The relationship of all these factors exists in this manner: The melting point of a given electrode is a function of its chemical composition and the fluidity of the molten drop is a function of its temperature, which in turn depends on the heat input. The surface tension of the molten metal is a function of the fluidity of the drop, also. Consequently, the molten drop of lowest surface tension will leave the tip of the electrode sooner than another drop of greater surface tension. Therefore, the most desirable bare electrode is the one that results in drops of low surface tension—and therefore of small volume. Small volume, on the other hand, means shorter intervals and hence a higher frequency.

There remains the time relationship between the formation periods and transfer periods to be examined. The longer the

formation period, the longer the time to preheat the base metal for the reception of the molten drop deposited on it. The longer the transfer period, the longer is the molten drop and the molten base metal exposed to atmospheric contamination. Consequently, a long formation period and brief transfer period are desirable. This requirement, however, can not be fulfilled with bare wire electrodes. The thin film covering of these wires, as mentioned before, is not sufficient to maintain an arc during the transfer period, so that during the partial short-circuit period, existing during the transfer period of each drop, the power input drops to about one-tenth of the arcing periods; thus the transfer is slowed down for lack of arc-conducting media.

Coated electrodes have a means to steady the arc during the transfer period, thus insuring a higher power input during the transfer period and thereby decreasing the time of contamination of the superheated drop and molten pool of the base metal. The coating of these electrodes consists merely of a wash of some kind of metallic oxide and perhaps only a coat of lime. The influence of this thin coating is exerted merely in cutting down the transfer period. Some of the metallic oxide coatings, however, alter the surface tension of the molten metal of the electrode and so influence the volume of each drop, thus changing the character of the electrode metal altogether. It is remarkable how the thin coating of an electrode changes its behavior through the arc, how it steadies the sequence of the drops and consequently how much denser is the resulting weld metal.

However, regardless of how much or how little the coating steadies the arc, there still remains a shorter or longer transfer period, during which the molten metal is exposed to air and does pick up oxygen and nitrogen, to an extent, that reduces the usefulness of the resultant weld metal to a degree where such metal is consigned today to only such work as has stresses of no importance. The only work of importance where bare or coated electrodes are still used is bridge repairing and reinforcing. It is hoped, however, that there, too, its use will soon be abolished.

Covered electrodes are the answer to various perplexities and contradictions that the welder expressed when formulating the

desirable features of arc welding electrodes. What was wanted may be summed up as follows:

(1) The electrode must permit the melting of a maximum amount of base metal per drop. This, however, meant a correspondingly large drop. Bare or coated electrodes can not fulfill this requirement. The best that may be done about them is to use electrodes that give a ratio between drop volume and volume of molten base metal per drop that is somewhat in favor of the base metal.

(2) The molten drop at the tip of the electrode, as well as the molten base metal should be protected from atmospheric contamination. In addition, the deposited metal should solidify under a protective coating of slag, which would prevent atmospheric contamination there and would permit the venting of occluded gases that might be liberated from the chilling metal.

Eventually, covered electrodes—often referred to as shielded arc electrodes—were developed. Today they are the only type of electrode used in important work. The Bureau of Engineering of the U. S. Navy specifies their use for all work of arc-welded construction.

Covered electrodes consist of a mild steel core of usually from 0.07 to 0.17 carbon content and covered with a matrix composed of 15 to 30 per cent volatiles, silica, and some other oxides, mostly reducers, all held together with a binder and put on the cores by the extrusion process. In other words, the covering is a refractory sleeve, which contains a certain amount of volatiles. The melting point of the refractory sleeve is somewhat higher than that of the core, so that the metal core melts sooner than the refractory sleeve. The result of this is that a small cup—a miniature crucible—forms at the end of the electrode. The importance of the presence of this miniature crucible is considerable—it changes the behavior of any electrode as compared with its bare performance. In the bare or thinly coated state the volume of each drop was entirely a function of the chemical analysis and the power input.

By changing the composition of the matrix, as well as its volume per unit length of electrode, the producer of covered electrodes is in the position to change the individual drop volume. Varying the composition of the matrix, *i.e.*, the melting point, and altering

its volume per unit length of electrode, the energy input, and consequently the arc potential, may be varied also until the desired effect is accomplished. The effect that is required is an elongation of the formation period resulting in relatively large volumes of molten base metal and the instanstaneous transfer of the completed drop.

The covering performs other equally important functions. Its volatiles furnish a neutral or reducing atmosphere that surrounds the arc, protecting the molten metal of the forming drop, the molten metal of the pool formed in the base metal, and finally the drop in transit through the arc. The non-volatile parts of the matrix, that are also transformed onto the work form a covering on the freshly deposited metal and, being directly exposed to atmosphere, solidify sooner than the metal underneath them, permitting the fused metal in the bead to breath and vent itself of its gases. The matrix of the covering, like the hearth of a steel furnace, may be acid, basic or neutral. It can not be either too acid or too basic, because if it is, then the resultant weld metal may suffer the consequences in the same manner as does a heat of steel turned out of a furnace having a hearth of improper composition.

Certain claims are made that the presence of certain reagents contained in the matrix help to further purify the metal. However, the opinion is ventured here that if the covering is capable of shutting off atmospheric contamination, it does a good deal of work. The reducing properties of the electric arc may be relied upon to purify the metal. There is time for the arc to actually purify the metal both in the drop and in the base metal pool—during the formation period. That it actually does purify the metal has been definitely proven. In the foregoing it was tried to illuminate the metallurgical influence of the covering.

The next property is in reference to the metallographic changes in the weld metal. When one speaks of weld metal, it is impossible to identify such with a typical grain structure as one finds in a steel casting, which a weld really is. A steel casting is either annealed or unannealed. Unannealed castings have a characteristic grain structure, which is entirely unlike that of the annealed steel. Of course, all annealed castings do not show the same grain structure, unless they are of approximately the same composition

and have received the same heat-treatment. Likewise, weld metal differs in grain structure depending on its composition and heat-treatment during its deposition.

The bulk of the weld metal gained from bare or thinly coated electrodes is of very low carbon content, because most of the carbon contained in the electrode is reduced by the arc. Since bare electrodes can not melt a great deal of base metal to alloy with, the individual beads are fairly high—they look like the segments of a cylinder. The smaller the amount of base metal that has been melted, the greater the height of the segment. If the weld is being built up layer upon layer—as it should be—the layer on top of the previous one will exert a certain amount of heat-treatment on the one immediately below it. The first bead put down has a grain structure of unannealed cast steel. The effect of the layer deposited upon it is equivalent to annealing. However, if the bead is too high, the annealing temperature zone will not penetrate to its full depth, and the bead will be but partially annealed. Covered electrodes deposit beads that appear to be shallow. While the ratio between width and height of bare electrode deposited beads is on average 1:½, that for covered electrodes is but 1:¼. The thinner each layer is, the better the heat-treatment it receives from its successor.

The grain structure of the covered electrode deposited weld metal—if the deposition was made layerwise—is composed of small, equi-axed grains. If the covering has been homogeneous, from a point of view of composition and density, and if the cohesion between matrix and core was uniform throughout the entire length of the electrode, permitting uniform performance, then the deposited metal is remarkably free of impurities, possesses a great density (7.8 c.g.s. system and over), has a high tensile strength, and a good elongation, coupled with a fine grain structure that greatly influences the impact strength of the material.

The physical properties of the weld metal depend upon the composition of the core and that of the matrix to a great extent. The strength of the deposited metal, being mild steel, depends to a great degree on its carbon content. The carbon in the deposited metal may find its way into it from two sources, namely, from the core and from the covering. The manufacturer of covered

electrodes may control, within limits, the composition, especially the carbon content of the weld metal, by varying the composition of the matrix only—without changing the core metal. If it is found desirable, from point of manipulation, to use as cores material of a certain carbon range but yielding weld metal of various physical characteristics, the manufacturer varies the composition of the matrix so that one batch yields a weld metal of somewhat higher carbon content than the other. For instance, one manufacturer, who uses almost exclusively .10-.14 per cent C. cores, is able to furnish covered electrodes as follows: Brand X has .10-.14 per cent C. cores, and is covered with matrix X-1. It yields a weld metal having a tensile strength averaging 65,000-68,000 p.s.i., with an elongation of over 25 per cent in two inches. This weld metal is relatively poor in carbon. Its covering is so designed that a certain amount of the carbon contained in the core is permitted to be reduced in the arc. Another brand, Z, which is also made using .10-.14 per cent C. cores, is covered with matrix Z-1, which permits retaining a large percentage of the carbon contained in the cores. This brand of electrodes yields a deposited metal with the following physical properties: Tensile strength, 70,000-75,000 p.s.i.; elongation 20 to 25 per cent in two inches.

The stresses introduced by electric arc welding are considerable, if certain precautions are not observed. The time element necessary to distribute the heat input of a single increment throughout the entire mass of the base metal is lacking, due to the swiftness of arc melting. The high temperature is restricted to the narrow band of metal of the weld zone. This narrow band of metal, wishing suddenly to expand, is prevented from free expansion by the practically unaffected mass of the base metal. There is, of course, some heat transmission gradually into the mass, but the high temperature band is cooling at a faster rate than is the warming up of the mass of base metal. The high temperature band, and especially the freshly deposited metal, has to contract in cooling, but is prevented from actually changing its dimension by the unyielding mass of base metal. The resistance set up by the unyielding base metal against the expanding weld zone band and the locked in stress due to the inability of the rapidly chilling weld zone band to contract freely, constitute the so-called residual

stresses. The value of this stress depends on two factors—first, the shape and relative position of the weld to other parts of the assembly that may limit all creeping of the metal or may permit of a certain amount of free movement, and secondly, the mean temperature of the weld zone during welding. If the latter is kept down to a minimum, the residual stresses may be kept low enough not to interfere with the safety of the completed structure.

According to the several codes, it is obligatory to stress-relieve certain welded assemblies, like pressure vessels, regardless of the method of welding pursued. The stress-relieving consists of heating the completely welded assembly to 1100-1200 degrees F., depending on its size and wall thickness, and keeping the assembly in the furnace at the specified temperature for a definite length of time, allowing it to cool in the still air of the furnace. The temperature of stress-relieving is not high enough to change the grain structure of the weld, or of the weld zone, or of the base metal. It serves only to redistribute the localized stresses by elevating the entire mass of the structure to a uniform temperature and exposing the mass to that temperature for a sufficiently long time to assure uniform stress distribution. It is possible, however, to minimize these stresses by keeping the mean temperature of the weld zone below an allowable maximum. This method may be accomplished by slowing down the pace of the work, requiring fairly long periods of waiting between depositions, to permit the cooling off of the mass. This procedure, of course, is time-consuming and expensive. However, the building of large stress-relieving furnaces, and the attending labor of moving the assembled structure into the furnace, and the bracing of the assembled structure in the furnace to prevent its warping or collapsing during stress-relieving, is also expensive. It is the opinion of the writer that when sufficient data are on hand, the bugaboo of stress-relieving will be handled not by means of a furnace operation only, which in many cases is apt to relocate the stresses instead of eliminating them, but by procedure control, in the same manner as it is done already in large field operations, such as welding hulls of ships, storage tanks of large dimensions, and other similar structures incapable of being housed for stress-relieving. In many cases the stress elimination of these structures is more or less

guess work, because the underlying principles are not well known and because there are not sufficient data on hand to enable the engineers to calculate. However, the trend of the times is such that in the near future more and more light will be thrown on the subject to help the art and science of arc welding to that scientific level it justly deserves.

EXPERIMENTS TO DETERMINE RELATIVE ELASTICITY OF MODIFICATIONS OF MAIN ENGINE CRANKSHAFTS.

By Wm. C. Stewart.

INTRODUCTION.

Twisting tests of an actual nine-throw Diesel engine crankshaft indicated that the member was considerably more elastic than was estimated from dimensions of it, using existing and heretofore reasonably satisfactory methods. This in turn resulted in bringing an exceedingly dangerous critical speed into the full speed range of the engine. There appeared to be two possible ways of explaining the disagreement of the twisting tests of the crankshaft with previous experience:

a. Substantial difference in the heat treatment of the material which may have decreased the modulus of rigidity beyond the normal values of previous experience.

b. Sufficient of a departure from the previously used geometric configuration of the crank, which would make the previously successfully used formulas for estimating elasticity inapplicable.

It appeared that the second condition was the most likely explanation of the discrepancies observed between the results of the twisting tests and the computed values for elasticity since it was hardly to be expected that heat-treatment or variable composition of the steel would appreciably affect the modulus of rigidity.

In order to investigate the influence of geometric configuration of the crank, model one-throw crankshafts of several different crankforms were prepared and tested. An equivalent hollow-bored cylinder, having the same diameter as the crankpin and journals of the model crankshafts, was likewise twisted.

Crankshaft models of three different compositions were tested in order to determine the influence of composition on the modulus of rigidity.

The effect of increasing the diameter of the bore on the relative rigidity was investigated by means of a considerably larger model than was used in connection with the experiments to show influence of crankform. Likewise, the design of the crankwebs of the model differed considerably from any of the modifications previously tested. In the experiments with this model no attempt was made to modify the crankform. Stiffness factors of the models were determined before and after increasing the diameter of the bore. As before comparison was made by means of an equivalent cylinder of the same composition and condition of heat treatment.

METHOD OF TEST.

Two carbon steel billets, 8 inches in diameter and 18 inches long, were selected for the preparation of the crankshaft model and reference cylinder. The crankshaft was machined directly from the billet without any additional forging operation. The reference cylinder was machined to a scale one-third that of the actual crankshaft diameter, namely, 9⅝ inches, whereas the crankshaft model was made to a scale of .331 due to a slight error in machining. Similar crankshaft models were prepared from 8-inch diameter billets of chrome vanadium and chrome molybdenum steel. As in the case with the carbon steel model, the alloy steel models were machined directly from the billets. No reference cylinders of the alloy steels were available, consequently all comparisons were made with the carbon steel cylinder.

Results of twisting tests of twelve modifications of the chrome molybdenum crankshaft designated No. 3 including similar test results of the plain carbon and chrome vanadium steel crankshaft models for the type VII crankform are presented in this paper. The latter models are designated crankshafts No. 1 and No. 2, respectively.

The crankshaft used to determine the effect of increasing the bore on the stiffness was received in the finished form. However, it is believed that the model was machined directly from a billet. This model was made to .888 the scale of the actual crankshaft, the latter member having 4½-inch diameter journals and crankpins.

Preliminary twisting tests of crankshaft models showed that satisfactory results could be obtained with the model mounted

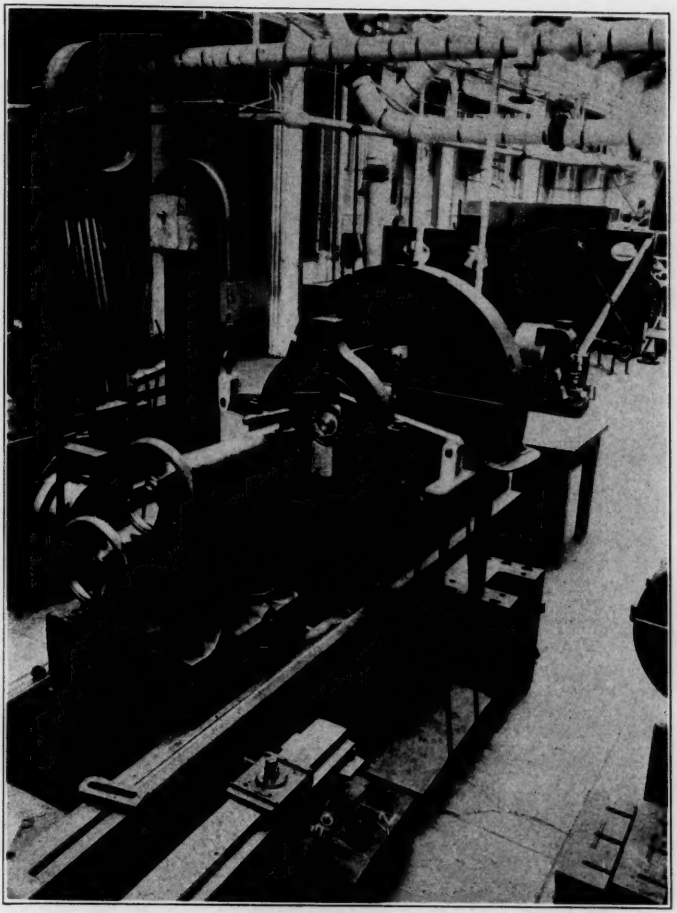


FIGURE 1.

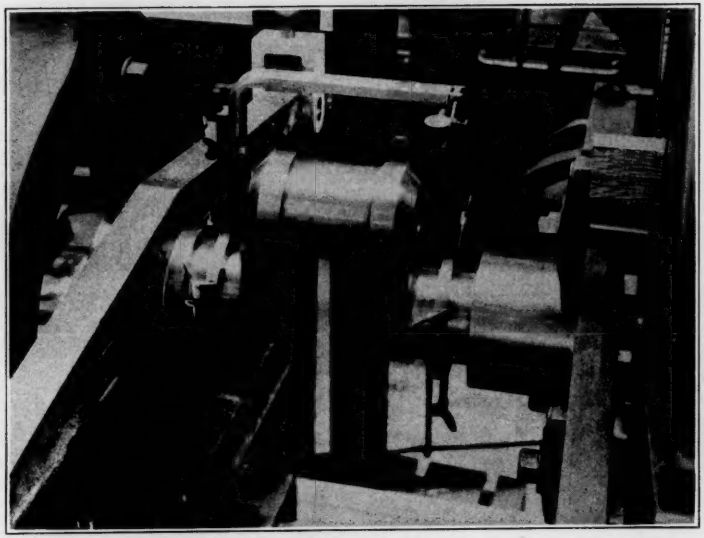


FIGURE 2.

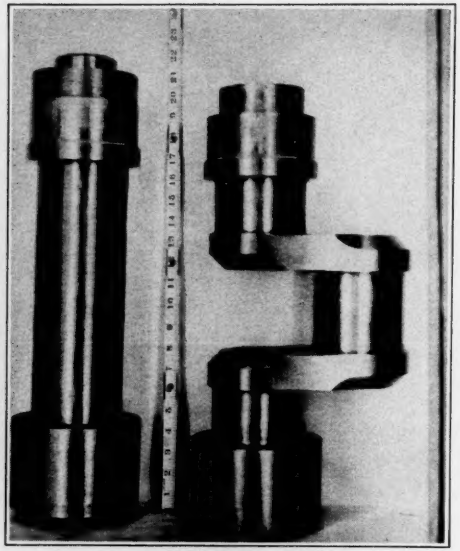


FIGURE 3.

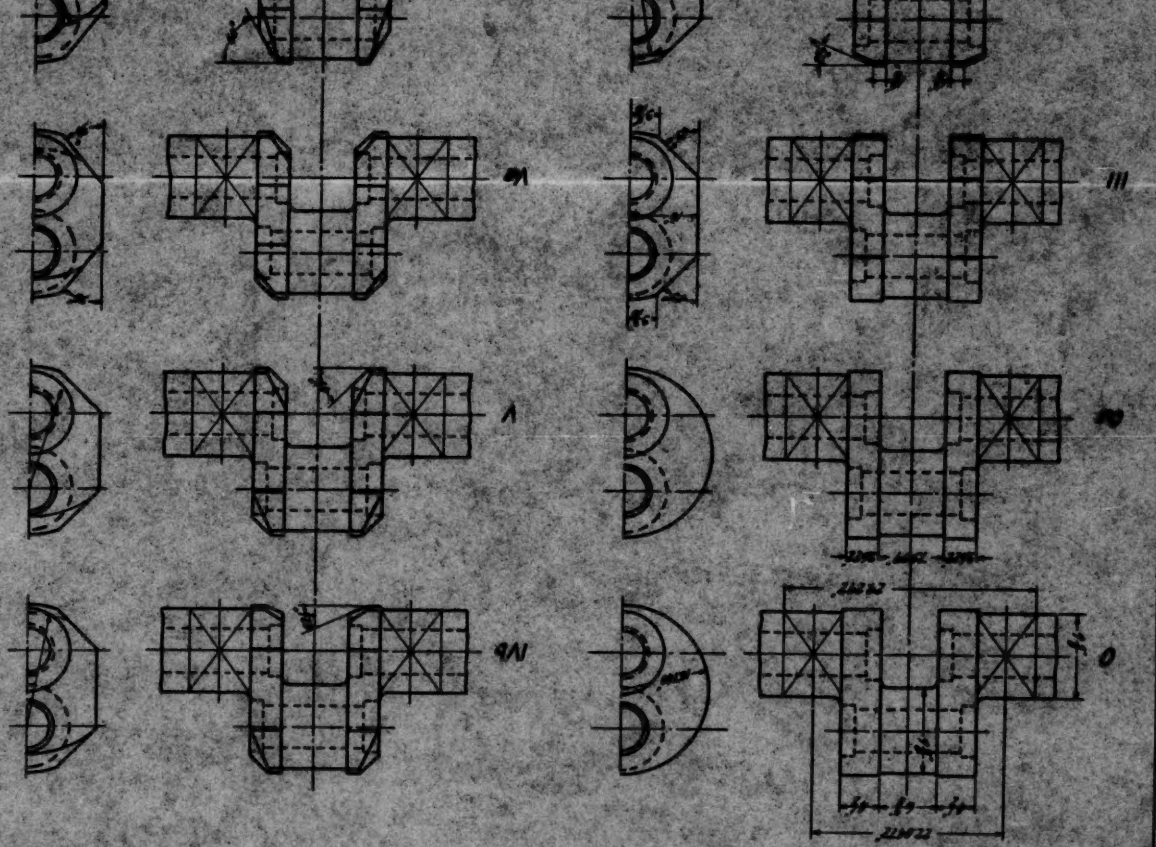
between the centers of a heavy lathe. A horizontal loading beam was keyed to the end of the crankshaft model so that the load was supported as near as possible to the tailstock center of the lathe. Torque was applied to the member by means of weights hung from the end of the loading beam. Movement of the opposite end of the model was prevented by means of a horizontal arm secured to the heavy flange of the crankshaft. In order to eliminate, as much as possible, reaction at the bearings, the load, in some cases, was applied to both ends of the loading beam after the manner illustrated in Figure 1. The illustration shows the loading end of the crankshaft provided with a large ball bearing. The ball bearing is supported on the 120-degree segments of the lathe, steady-rest with an additional supporting block beneath. Lugs attached to the face plate hold the shaft on the lathe center. The tailstock center can be used to centrally locate the shaft. After the steady-rest supports are adjusted the center is withdrawn. Clearance between the balls and the race of the bearing prevents any considerable horizontal restraint of the member. In the case of the reference cylinder the ends of the bore were provided with centered steel plugs for mounting the shaft between the lathe centers. Some tests of crankshaft models were made with the model mounted between both lathe centers, whereas in other tests the ball bearing support was used.

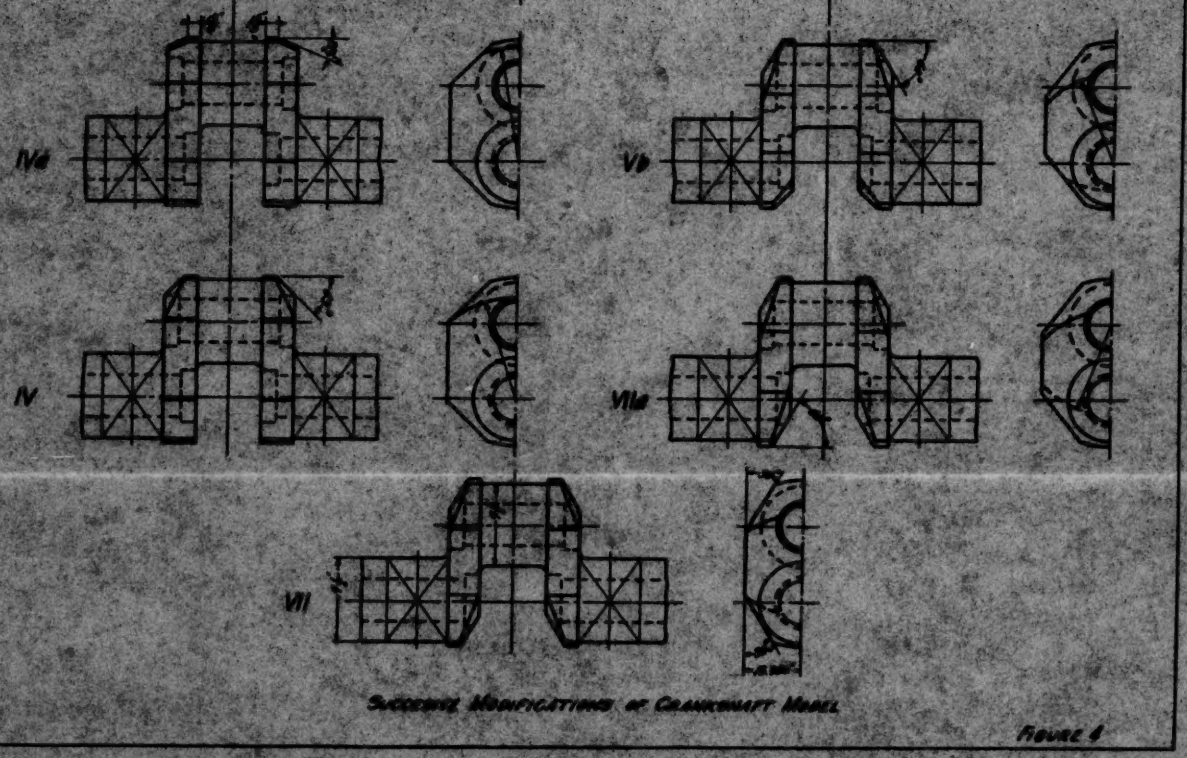
MEASUREMENT OF DEFLECTION IN THE CRANKSHAFT MODELS DUE TO TORQUE.

As a result of preliminary tests it was evident that methods of determining the relative deflection between midpoints of the crankshaft journals were unsatisfactory if the indicators were mounted on the foundation and actuated by arms attached to the crankshaft journals. Deformation of the supporting structure introduced errors into the readings obtained by these methods. Torque-deflection diagrams obtained in this manner showed decided hysteresis loops. In many cases the results were reproducible since errors of the same magnitude were introduced into the readings of each test. However, in general, the results were unreliable. Accordingly, it was considered necessary to develop a method that would

be independent of the structure supporting the member under test. It appeared that torsion rings mounted on the journals and arranged so that an arm from one ring made contact with a sensitive indicator secured to the mating ring would be independent of any movement encountered in the supporting structure. Accordingly, deflection between midpoints of the crankshaft journals resulting from torque applied to the end of the shaft was determined by means of torsion rings and a "Last Word" dial indicator. These rings were secured to the journals by means of pointed set screws mounted in the rings at intervals of 120 degrees. Figure 2 presents an illustration showing the rings mounted on a crankshaft model. The rings are split to facilitate mounting on the journals. The ring sections are tongue and grooved, being held together by set screws. The dial indicator is secured to the vertical arm of one of the rings in such a manner that the ball contact of the indicator touches the horizontal arm of the mating ring in a definite location. Fixtures are used to center the rings both with respect to midpoint of the journals and the center of the shaft. In this way the rings can be remounted in precisely the same location if removed from the shaft. Horizontal extension arms of various lengths are available so that the span can be varied. Likewise, provision is made for adjusting the radius of the arc, that is, the distance between the center of the shaft and the ball contact of the indicator. The same torsion rings were used in testing the crankshaft models and the reference cylinders. Likewise, the same rings fitted with inserts to reduce the diameter were used in testing the .750-inch diameter torsion specimens. These inserts support the three-point set screws, thus making the assembly more rigid when used in connection with relatively small diameter test bars. Obviously, this method of determining deflection as a result of torque is not applicable in cases where large angular deflections are encountered. However, the method was admirably adapted for the character of tests described in this paper.

Chemical analyses, tensile and static torsion tests were made of specimens prepared from 1⅛-inch square billets representing the chrome vanadium and chrome molybdenum steel crankshaft models. In the case of the carbon steel crankshaft model designated No. 1 and the reference cylinder, similar tests were made of specimens





Successive Modifications of Crankshaft Model

Figure 4

prepared from an 8-inch diameter forging representing the same heat of material. No test specimens were available representing crankshaft model No. 4. However, a sample for chemical analysis was obtained from drillings taken from the end of the shaft.

RESULTS OF TWISTING TESTS.

The series of experiments herein described were the result of two different investigations to determine the relative stiffness of crankshaft models as affected by change in crankweb form and diameter of bore. The first experiments were made using a one-throw crankshaft model of approximately one-third scale machined from a chrome molybdenum steel forging. The model was tested following each of eleven modifications in crankform and after one reduction in journal and crankpin diameter. In the case of one form of crankweb, namely, type VII, models of chrome vanadium and carbon steel were tested in addition to the chrome molybdenum steel model. Comparison was made with an equivalent hollow-bored cylinder of carbon steel. The models were subjected to a maximum torque of 20,100 inch-pounds in increments of approximately 1000 inch-pounds. The load was removed in the same increments as applied in order to ascertain if any appreciable hysteresis was encountered and to determine if the indicator returned to zero reading. Reference to the sketches of Figure 4 will show the various crank configurations that were investigated.

The second series of tests was undertaken with the view to determining the stiffness ratio for crankshaft models as affected by increase in bore diameter. In these experiments no attempt was made to alter the form of the crankwebs. It will be noted from the illustration, Figure 3, that the crankweb design for this model differs considerably from any of the models shown in Figure 4. As in the previous experiments comparison was made by means of measurements obtained with an equivalent hollow cylinder prepared from a steel forging of the same approximate composition as the crankshaft model. As shown in the illustration the model consisted of one crankthrow and was made .888 the size of the actual six-throw crankshaft. A maximum torque of 47,700 inch-pounds in thirteen increments of approximately 3600

inch-pounds was applied to this model. The load was removed in the same order as applied in order to check the deflection readings and to ascertain if the dial indicator returned to zero.

RESULTS OF TWISTING TESTS OF CRANKSHAFT MODELS FOLLOWING MODIFICATIONS IN CRANKFORM.

Results of twisting tests of the .331-scale crankshaft models are presented in Table I. The data for the various crankshaft modifications as shown in this table represent results as obtained for the corresponding models of the same type number shown in Figure 4.

RESULTS OF TWISTING TESTS OF CRANKSHAFT MODELS AS AFFECTED BY MODIFICATION IN CRANKWEB FORM

Model	Type	Successive wt. loss, lbs.	Stiffness Factor	Total twist between journal centers for torque of 20100 in.lb.		Twist/ linear inch of shaft, radian	Twist for 20100 in.lb., Radian				
				Inch	Radian		Journals	Pin	Pin and journals	Webs	Webs per linear inch
3	0	0	1.369	.01019	.001035	.00082	.000411	.000399	.000810	.000225	.000076
3	0a	3.40	1.232	.01085	.001102	"	.000436	.000455	.000891	.000211	.000088
3	III	2.66	1.113	.01204	.001223	"	"	"	"	.000332	.000138
3	IVa	.04	1.107	.01210	.001229	"	"	"	"	.000338	.000141
3	IV	.30	1.107	.01210	.001229	"	"	"	"	.000338	.000141
3	IVb	.22	1.113	.01204	.001223	"	"	"	"	.000332	.000138
3	V	.23	1.103	.01215	.001234	"	"	"	"	.000343	.000143
3	Va	.95	1.085	.01234	.001254	"	"	"	"	.000363	.000151
3	Vb	.32	1.072	.01250	.001270	"	"	"	"	.000379	.000158
3	VIIa	.50	1.058	.01278	.001298	"	"	"	"	.000407	.000170
3	VIIb	1.98	.939	.01430	.001453	"	"	"	"	.000562	.000235
3	VII	.08	.915	.01468	.001491	.00092	.000460	.000480	.000940	.000551	.000230
1	VII		.920	.01461	.001484	"	"	"	"	.000544	.000227
2	VII		.907	.01480	.001504	"	"	"	"	.000564	.000235
	VIII		1.000	.013678	.0013896	.00087					

TABLE I.

As previously mentioned models No. 1, No. 2 and No. 3 refer to crankshafts of carbon, chrome vanadium and chrome molybdenum steel, respectively. Type VIII indicates the carbon steel equivalent

cylinder by means of which all comparisons were made. Column 4 gives the stiffness factors of the various modifications based on unit stiffness for the reference cylinder. Column 3 gives the successive loss in weight of the models as a result of the changes in crankform and journal diameter. The twist in the journals and crankpin of the models at the maximum torque is determined from the twist in an equivalent length of the reference cylinder at the same torque. The total twist in the crankshaft less the twist in the journals and crankpin is attributed to twist in the crankwebs. Dimensional data relative to the full size crankshaft are presented in Table II.

DIMENSIONAL DATA IN REGARD TO FULL SIZE CRANKSHAFT

Model No.	0 *	0a	III	IVa	IV	IVb	V	Va	Vb	VIIa	VIIb	VII
Width of Crankweb, in.	18.74	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	16.0	13.39	13.39
Chamfering angle, deg.	0	0	45	45	45	45	45	52	52	52	60	60
Bevelling angle, deg.: Pin	0	0	0	22½	45	45	45	45	60	60	60	60
Journals	0	0	0	0	0	22½	45	45	45	60	60	60

* 18.74" dia. webs - 4½" thick arbitrarily chosen.

All angles are given in degrees from a horizontal line.
By chamfering is meant reduction in width of crankweb by cutting off all four corners of each web.
Bevelling refers to reduction in thickness of the web-journal and crankpin ends being bevelled in separate steps.
Dimensions refer to full-size shaft and all shapes tested had 9-5/8" diameter crankpins and journals, except model VII, which represents original MAN design with 9-1/8" diameter pins and journals.

Model crankshaft types 0 to VII, inclusive, were made to 0.331 scale.
Cylinder (type VIII) was made to 0.333 scale.

TABLE II.

CHEMICAL COMPOSITION AND PHYSICAL PROPERTIES OF MATERIALS REPRESENTING CRANKSHAFT MODELS.

Tables III and IV give the chemical composition and physical properties, respectively, as obtained for forgings of approximately the same composition as the crankshaft models. The results of analyses presented for crankshaft No. 4 and the corresponding equivalent cylinder were obtained from samples drilled from the actual sections.

CHEMICAL COMPOSITION OF MATERIAL
All values in per cent

Material	Representing	Original size of stock	C	Mn	P	S	Si	Ni	Cr	Cu	V	Mo
Chrome-Molybdenum Steel	Crankshaft Model #3	1-1/8" sq.bar	.44	.75	.017	.012	.10	.69	.95		.30	.196
Chrome-Vanadium	Crankshaft Model #2	1-1/8" sq.bar	.38	.56	.016	.022	.10	.33	.97		.26	
Carbon Steel	Crankshaft Model #1 Ref:Cylinder Type VIII	8" dia.billet	.49	.53	.015	.031	.24	None	.053	.01		
Carbon Steel	Crankshaft Model #4		.27	0.42	.010	.027	.16	Less than .05	.10		None	None
Carbon steel	Reference cyl.corresponding to #4 crankshaft model		.25	.46	.038	.033	.17	Less than .05	.02		None	None

TABLE III.

STATIC TENSION, CHARPY IMPACT AND HARDNESS TEST RESULTS

Material	Condition	Original size of Stock	Tensile Strength	Johnson's Limit	Proof Stress	Elastic Limit	Proportional Limit	Reduction of area	Elongation in 2 in.	Charpy Impact value ft.lb	Brinell Hardness No.
			Pound per square inch					Per cent			
Chrome-Molybdenum Steel	As rec'd.	1 1/8" sq. Bar	148600	80750	81880	70750	61250	42.8	14.7	16.8	291
Chrome-Vanadium Steel	As rec'd.	1 1/8" sq. Bar	120000	82880	84750	82500	63000	59.1	17.0	26.4	238
Carbon Steel	Quenched and Drawn	8" Diam. Billet	110700	60750	61500	55500	52080	49.5	24.2	5.5	196

TABLE IV.

METHOD OF LOADING CRANKSHAFT MODELS AS AFFECTING TEST RESULTS.

A series of twisting tests was made in order to determine if the manner of loading the beam or the method of supporting the model appreciably affected the test results. For this purpose model type VII for crankshafts No. 1, No. 2 and No. 3 were selected. Three different methods of support for the crankshaft models were tried:

(1) Both ends supported on lathe centers.

(2) Loading end fitted with ball bearing supported by steady-rest, opposite end on lathe center.

(3) Loading end fitted with ball bearing supported by steady-rest, opposite end on lathe center, crankpin supported on ball bearing rollers.

For each of the three methods of supporting the models two different loading procedures were tried. First, the entire load necessary to produce a torque of 20,100 inch-pounds was applied to one end of the loading beam in increments equivalent to approximately 1000 inch-pounds. Following this procedure the same increments of load were applied alternately to the opposite ends of the beam after the manner illustrated in Figure 1. Results of these tests for the three type VII crankshafts and the reference cylinder are presented in Table V. The slope values as taken from the torque-deflection diagrams are used as a means of comparing the different values.

COMPARISON OF LOADING METHODS

Shaft Model	Shaft Type	*	Supported on Lathe Centers		One end on Lathe Center, other end in ball bearing		One end supported on lathe center, opposite end and center supported on ball bearings	
			Loaded on one side	Loaded on both sides	Loaded on one side	Loaded on both sides	Loaded on one side	Loaded on both sides
1	VII	a	.1240	.1242	.1243	.1245	.1252	.1235
		b	.01496	.01498	.01500	.01502	.01510	.01490
		c	.001520	.001522	.001524	.001526	.001534	.001514
2	VII	a	.1230	.1247	.1242	.1257	.1238	.1258
		b	.01484	.01504	.01498	.01516	.01494	.01518
		c	.001507	.001528	.001522	.001540	.001518	.001542
3	VII	a	.1242	.1240	.1228	.1240	.1235	.1240
		b	.01498	.01496	.01482	.01496	.01490	.01496
		c	.001522	.001520	.001506	.001520	.001514	.001520
Hollow Cylinder	VIII	a	.1121	.1116	.1128	.1116		
		b	.01352	.01346	.01360	.01346		
		c	.001374	.001367	.001382	.001367		

All slopes were calculated from actual deflection at 20100 in.lb.torque.

* (a = Slope
(b = Twist over 7.434 in. test length with 9.843 in. radius,in.
(c = Same as (b) expressed in radians.

TABLE V.

TWISTING TESTS TO DETERMINE EFFECT OF INCREASING DIAMETER OF BORE OF CRANKSHAFT MODELS.

Results of twisting tests of a crankshaft model having 4-inch diameter journals and pin are given in Table VI. Both crankshaft and reference cylinder were tested with the original bore of 1.305-inch diameter. Following these tests the bore of each member was increased to 2.277-inch diameter and the tests repeated. The stiffness factors for the crankshaft models and the modulii of rigidity

for the cylinders are shown in the columns to the right of the table. This crankshaft model was .888 actual size. From the results of test of crankshaft No. 4 it will be noted that 59 per cent of the total deflection of the member is attributable to the cylindrical portion whereas 41 per cent is due to twisting in the webs. In the case of crankshaft No. 4*a* having the 2.277-inch diameter bore, 52 per cent of the total twist occurred in the journals and pin, and 48 per cent in the webs. Therefore, the increased bore had relatively greater effect on reducing the stiffness of the webs than it had on the pin and journals as compared with crankshaft No. 4. Consequently, the stiffness ratio for the crankshaft with the 2.277-inch diameter bore is less than for the original form. It appears that the stiffness of the member could be increased somewhat by reducing the diameter of the bore in the webs. Cylinder No. 4*a* with the enlarged bore showed a somewhat higher modulus value than for the original form. The former value was in better agreement with the values as ordinarily obtained for relatively small dense bars. Removing additional material from near the center of the cylinder probably resulted in a section relatively more dense. Since the elastic modulus seems to depend more on the density of the material than on any other physical property, it might be expected that the cylinder in the final form would show a slightly higher value for the shearing modulus.

RESULTS OF STATIC TORSION TESTS.

Results of static torsion tests of specimens prepared from 1⅛-inch square billets representing the chrome vanadium and chrome molybdenum crankshaft models are shown in the two upper columns of Table VII. The results shown for the 8-inch diameter billet represents the same heat of material as the carbon steel crankshaft. The modulii shown in column 6 were obtained from various torque values as indicated in column 8. Different types of indicator dials were used as shown in column 11. Similarly, the radius (distance between center of the specimen to indicator contact) and the test length were varied. These variations were introduced in order to subject the torsion meter to as many different conditions as possible. The torsional modulii as obtained for the reference cylinder are in excellent agreement with the corresponding values for the 1⅛-inch square alloy steel billets. The shearing modulii were computed from the formula.

$$\theta = \frac{Nt\,l}{GI_p} \text{ where}$$

θ = twist in radians

Mt = torque, inch pounds

l = distance between gauge points

G = modulus of rigidity

I_p = Polar moment of inertia of cylinder

$$\theta = \frac{\text{deflection dial indicator (inch)}}{\text{radius (center of shaft to dial contact, (inches))}}$$

TORSION TEST DATA FOR MODEL CRANKSHAFT AND HOLLOW CYLINDER

Model	Outside Dia. in.	Bore, in.	Avg. slope *	Avg. total twist between journal centers (9.332") for torque of 47700 in.lbs.	Twist for 47700 in.lbs., Radian				Modulus of rigidity, lb./sq.in.	Stiffness Ratio
					Per linear inch of shaft	Journals	Pin	Pin and Journals	Webs	
				inch / Radian						
Cylinder #4	4.000	1.305	.1566	.0149	.0015148	.0001623	-	-	-	11.823x10^6	
Cylinder #4a	4.000	2.277	.1688	.0160	.0016236	.0001749	-	-	-	12.120x10^6	
Crankshaft #4	4.000	1.305	.1776	.0169	.0017175	.0001840	.0003966	.0006130	.0010096	.0007079	-	0.862
Crankshaft #4a	4.000	2.277	.2145	.0205	.0020793	.0002228	.0004775	.0005606	.0010981	.0009214	-	0.785

* Scale (Abscissas - 1 inch per 1000 in.lbs.
(Ordinates - 1 inch per .0020 inch deflection = .00020278 radian.

$\theta = \frac{\text{Arc}}{\text{Radius}} = \frac{\text{Deflection (inch)}}{9.863}$

TABLE VI.

STATIC TORSION AND SHEAR TEST RESULTS

1	2	3	4	5	6	7	8	9	10	11	12	13	14
Specimen Number	Original size of Stock	Average maximum Torsional strength, lb/sq.in.	Average Torsional Proportional limit, lb./sq.in.	Average Single Shearing strength lb./sq.in	Modulus of Rigidity, lb./sq.in.	Fiber Stress at Torque, in Col.8, lb./sq.in.	Torque in.lb.	Twist Inch	Twist Radian	Dial	Radius in.	Test length in.	Specimen diameter in.
					CHROME MOLYBDENUM STEEL								
AMC 100	1 1/8" Sq. Bar	89300	47100	101500	11.926 x 10^6	13150	1090	.21525	.021868	Ames	9.843	7.434	.750
" "					11.743 "	18560	1538	.2742	.031337	"	8.750	"	"
" "					11.760 "	22200	1839	.2245	.037417	"	6	"	"
" "					11.599 "	20370	1688	.20892	.034820	Shore	6	"	"
" "					11.667 "	24040	1992	.16343	.040852	Ames	4	"	"
" "					11.562 "	22200	1839	.15222	.038055	Shore	4	"	"
" "					11.762 "	25880	2144	.1056	.017600	Ames	6	3	"
" "					11.679 "	25880	2144	.0709	.017725	"	4	3	"
" 150					11.740 "	25880	2144	.1058	.017633	"	6	3	"
" "					11.646 "	25880	2144	.0711	.017775	"	4	3	"
				Average	11.708 "								
				*	±.078 "								
					CHROME VANADIUM STEEL								
AMC 50	1 1/8" Sq. Bar	78900	44700	85500	12.110 x 10^6	14070	1166	.22675	.023077	Ames	9.843	7.434	.750
" 1					11.908 "	20770	1688	.2035	.033917	"	6	"	"
" "					12.063 "	25880	2144	.1701	.042525	"	4	"	"
" "					11.943 "	25880	2144	.1040	.017333	"	6	3	"
" "					11.829 "	25880	2144	.0700	.017500	"	4	3	"
				Average	11.971 "								
				*	± .093 "								
					CARBON STEEL								
SBB 31001	8" diam. Billet	76100	39400		11.778 x 10^6	1730	10000	.00679	.000689	Last Ford	9.843	7.434	Hollow
" "					11.756 "	3480	20100	.01368	.001389	"	"	"	Cylinder
" "					11.776 "	4240	24550	.01666	.001693	"	"	"	3 166 OD,
" "					11.777 "	5280	30560	.02074	.002107	"	"	"	1.640 ID
				Average	11.772 "								
				*	± .008								

* Mean departure from the average

TABLE VII.

COMPARISON OF VALUES FOR SHEARING MODULUS AS REPORTED BY VARIOUS INVESTIGATORS

Investigator	Metal	Torsional Modulus, lb. per sq.inch	Remarks
Reuleaux		2/5 of Young's modulus	No experimental results are given. No reference to sources.
Cloud	Steel	12600000	Cloud calculated this value from formulas of Reuleaux. No direct determinations of G were given.
Benjamin & French	Tempered Steel	12000000 to 18000000	Determined by compressing springs in Riehle compression testing machine. Their method was probably inaccurate as this variation has never been indicated by any other investigators.
A.A. Cary	Steel	12000000 to 13000000	Subjected straight sections to torsion
J.K.Wood	0.85% C steel in form of music wire	11800000 to 12100000	Used the following three methods: 1. Subjected sections to torsion. 2. Subjected springs to deflection. 3. By torsional oscillation of a system consisting of the wire to which a weight was attached.
E.W.Stewart	Many types of steel	11500000	Determined by deflection of springs. In over a hundred tests on drawn steel wire the variation in modulus was within plus or minus 5%
	Phosphor Bronze	6250000	
	Brass	5000000	
	Monel Metal	9200000	
W.P. Wood	Hard drawn and tempered steels	10009000 to 11865000 Grand avg. 11454000	Deflection of helical springs loaded with dead weights
U.S.Naval Engineering Experiment Station	Cr-Mo steel from 1-1/8" sq.rods	11708000	Determined by subjecting cylindrical sections to torsion.
	Cr-V steel from 1-1/8" sq.rods	11971000	
	.49% C steel from 8" diam. billet	11772000	
	.25% C steel hollow cyl.,4.00" O.D. and 1.305" I.D.	11823000	
	.25% C Steel hollow cyl.,4.000" O.D. and 2.277" I.D.	12120000	
	Average ——	11879000	
	Mean departure from average ——	± 133000	
Values Generally Used	Steel	12000000	
	Phosphor Bronze	7000000	
	Brass	5000000	
	Monel Metal	9000000	

In connection with values obtained by above investigators, refer to "Torsional Modulus of Carbon Steel, Phosphor Bronze, Brass and Monel Metal" by W.P. Wood, Transactions, American Society for Steel Treating, Vol. 15, 1929.

TABLE VIII.

Values for the modulus of rigidity, particularly of steel, as reported by various investigators are presented in Table VIII. The corresponding results as determined from the reference cylinders and .750-inch diameter specimens presented in this paper are included for convenience of comparison. It will be noted that these results are in good agreement with the values reported by other investigators.

CONCLUSIONS.

It is believed that the method described for determining the relative stiffness of crankshaft models will give reliable results. It appears that similar methods may be developed for use in connection with similar tests of multi-throw crankshaft models or actual crankshafts. In these cases supporting rollers would be required to prevent bending of the member. Deflection measurement could be taken from journal to journal over the entire length of the shaft by means of torsion rings and dial indicators. It appears that a test assembly of this kind would be independent of the supporting structure, a condition much to be desired in connection with the twisting of crankshafts.

ELECTROMAGNETIC WAVES.

By Wm. H. Crew.

A redeeming feature in the startling growth of scientific knowledge in the past three centuries is the ever simplifying relation between its various parts. Time was when the propagation of light and of sound were phenomena in strictly different fields of knowledge, yet it is now well known that they have a common ancestry in the great family of waves,[1] and that they are first cousins to waves in liquids and in solids. Since the middle of the last century, a number of optical and electrical phenomena have been discovered, which in time were found to be closely related—indeed, to be members of the magnificent clan of electromagnetic waves; and, although in external appearance they differ one from the other, yet like "Mrs. O'Grady and the Colonel's Lady" they are nevertheless "sisters under the skin."

Let us trace the series of ideas that led scholars of the eighteenth and nineteenth centuries to the realization that light partakes of a kind of wave motion; and then see how certain other phenomena such as radio, X-rays, gamma rays, etc., were found to be component parts of one great electromagnetic spectrum of which the family tree is shown in Figure 1. Finally we shall digress but slightly from the electromagnetic spectrum to consider the wave nature of matter—a concept which has but recently found its way into the scheme of things.

The Wave Nature of Light: Following common practice let us use the term radiation to refer to any part of the electromagnetic spectrum, and reserve "light" to denote that part of the spectrum perceived by the human eye.

About the time that Sir Christopher Wren was reconstructing London after the great fire of 1666, Newton made the first analytical study of light by placing a prism of glass in the path of a ray from the sun. He observed the ray to be separated into various

[1] "Waves," Journal of American Society of Naval Engineers, August, 1932.

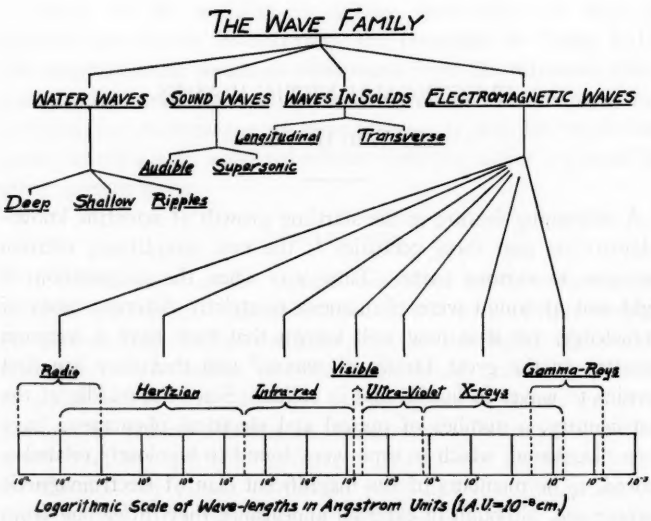


FIGURE 1.

colors, which he considered to be excited in the eye by corpuscles of light traveling at very high speed in air. When such particles enter a medium denser than air, for example, glass or water, he supposed their speed to be *increased*—those which were speeded up the most giving us the sensation of blue light and those speeded up the least rendering the impression of red. This explanation by Newton had the distinction of being consistent with the well known law of refraction discovered in 1621 by the Dutch mathematician and geodicist, Willebrord Snell.

Newton, however, failed to take cognizance of what might have appeared at the time as a rather insignificant observation by Grimaldi (1618-1663) in Italy. Grimaldi noticed that the shadows of opaque objects do not end abruptly, but show a curious gradiation at the edges, a fact which suggested to the great physicist, Christian Huygens (1629-1695), that light is propagated not as a corpuscle but as a wave-impulse. Huygens, by the way, was a member of the celebrated quadumvirate of contemporaneous bachelors, the others being the illustrious Newton, Descartes, and

Robert Hooke, all worthy contributors to the scientific knowledge of the seventeenth century.

It was immediately realized that if light is of an undulatory nature, then, in order to explain Newton's prismatic colors, it is necessary to assume that light waves are *retarded* by media denser than air, rather than accelerated as Newton had supposed; and that blue light is so retarded more than red light. Thus Huygens' wave theory led to an inference regarding the velocity of light in dense media which was in direct opposition to that drawn from the corpuscular theory. It may be added here that, for over one hundred years after the time of Newton and Huygens, the scientific world gave greater credence to the corpuscular theory largely because Newton, its champion, bore a greater reputation than his Dutch contemporary. The facts of the case in this matter were unknown until about 1850 when the very careful and precise measurements of Foucault and later of Michelson[2] showed that light is retarded when entering a dense medium. The wave hypothesis became a generally accepted theory of light about the time that Lord Nelson was establishing England's supremacy on the seas, for it was then that the eminent physician, philosopher, and scientist—"the last man who knew everything"—Thomas Young, did his famous experiment of interference with a "double slit." The observations of this experiment harmonized beautifully with the idea of waves, but found no ready explanation on the basis of the corpuscular theory, consequently the wave theory led a rather complacent life for nearly one hundred years.

Toward the end of the nineteenth century, however, a brilliant young German physicist, Heindrich Hertz, is alleged to have made the remark: "The wave theory of light has come to stay." Hertz died at the early age of thirty-six little realizing that a casual observation made by him while performing an experiment was to open up the fields of photoelectricity which today remains the most prominent optical effect unexplained by the wave theory of light, but which is satisfied by a revised corpuscular theory. Here, then, are two prominent radiation phenomena, the photoelectric effect and interference, each championed by a reputable and distinct

[2] "Velocity of Light," E. W. Thomson, U. S. Naval Institute Proceedings, *56*, 42 (January, 1930).

theory of light, yet both wanting common explanation. Sir Oliver Lodge likens these two theories to the shark and the tiger, "each supreme in its own element but helpless in that of the other."

For the present, however, let us pass over this incongruity and leave the theory of light as Hertz saw it at the end of the nineteenth century firmly seated on the wave hypothesis.

Our next step is to climb into the higher branches of this family tree and consider some experiences closely allied to light. More than a century after Newton's observation of the prismatic colors, Sir William Herschel discovered (1799) that when he placed a sensitive thermometer just beyond the red end of the spectrum it recorded more radiant heat than when in the red itself. He then showed that these heat radiations are capable of reflection, refraction, and interference, and are therefore merely a continuation of the visible spectrum in the direction of longer wave-lengths. Whereas light waves range in length from about a thirty-thousandth to one sixty-thousandth of an inch, these so-called infra-red rays are known to be as long as one-fiftieth of an inch.

In the year following Herschel's discovery it was observed by J. W. Ritter, then of Jena, that there are also radiations beyond the violet end of the visible spectrum which will blacken silver chloride. In modern spectroscopy these ultra-violet rays are readily recorded on photographic plates and in extreme cases are found to be less than one-millionth of an inch in length.

It is interesting to note in passing that certain substances, such as glass, which are transparent to visible light are totally opaque to ultra-violet radiation; so that people who live in "glass houses" may not only "not throw stones," but they may also be denied a Palm Beach tan. Thin layers of certain gelatinous substances may also be perfectly opaque to light yet quite transparent to the long infra-red waves. This is somewhat true of canvas, as may be appreciated by those who have slept in a tent and have been awakened by the heat of the morning sun without the realization of much daylight within.

No one of the present day and age experiences any difficulty in associating radio transmission with waves, yet it may not be common knowledge to all that these waves actually are subject to the well known optical laws of refraction, reflection, interference,

and polarization. Furthermore, by long experimentation requiring the essence of skill and technique, Nichols and Tear of Cleveland in 1923 produced radio waves of shorter wave-length than the longest infra-red waves and showed the two to be identical in the overlapping regions.

In 1896, the German physicist, Roentgen, discovered an invisible ray which would blacken a photographic plate and also penetrate many bodies which are opaque to other radiations. Because of its unknown character it was, at the time, named the X-ray. In the year 1912, Laue, Friedrick, and Knipping found that these X-rays possessed the optical characteristics of interference and diffraction; and later their wave-lengths were accurately measured by Sir William Bragg at the University of Manchester, and found to range from more than one ten-millionth to less than one-billionth of an inch. It was thus shown that the longer X-rays overlap the region of the shorter ultra-violet radiations.

Other rays, which are sponstaneously emitted from radioactive substances such as thorium and radium, also show optical properties. They have been called "gamma rays" and are estimated to have a wave-length range from over one hundred-millionth to under one ten-billionth part of an inch.

Still another type of ray, whose origin is unknown, is the cosmic ray. This has received front-page popularity on many recent occasions; but like many other front-page celebrities, its character is doubtful; some hail it as a new wave-phenomenon, while others believe it to be corpuscular in nature. Suffice it to say that the cosmic ray, if indeed it is a wave, has a wave-length many times shorter than the shortest gamma ray.

In the foregoing discussion it has been convenient to consider these radiations under six different headings—different because they produce different effects; some, for example, give the sensation of light; others of heat; some produce sunburn, others may be rendered audible by a radio apparatus. There are several reasons for believing that these radiations which exhibit such a wide variety of characteristics are fundamentally the same phenomenon. First of all, experiments indicate that these radiations all have wave characteristics such as interference and diffraction; secondly, in spectral regions where one type of radiation overlaps

another the two concerned are found to be identical; and thirdly, overlapping has been observed between every pair of adjacent types of radiation. This situation may be likened to the sounds from a string quartet, each instrument having, in general, a characteristic pitch and quality, the higher frequencies of the cello being identical with the lower frequencies of the viola, and the latter in turn overlapping the range of the second violin, which likewise overlaps with the first. The overall range of the quartet might appropriately be referred to as the sonic spectrum of stringed instruments, just as we speak of the spectrum of electromagnetic radiations.

In Figure 2, the various types of radiation are listed in tabular form with their approximate limits of wave-length in centimeters, and the names and dates appropriate to them. This tabular array of waves is known as the electromagnetic spectrum. It now remains to discover why it is called electromagnetic.

THE ELECTROMAGNETIC SPECTRUM.

Type of Radiation.	Range of Wave-lengths in centimeters.	Investigators and associated dates.
Radio	2.0×10^{6} to 50	Marconi; 1899
Hertzian	1.0×10^{3} to 3.0×10^{-2}	Hertz; 1887 Nichols and Tear; 1923
Infra-red	3.0×10^{-2} to 7.8×10^{-5}	Sir Wm. Herschel; 1799
Visible	7.8×10^{-5} to 4.0×10^{-5}	Newton; 1672
Ultra-violet	4.0×10^{-5} to 1.5×10^{-6}	Ritter; 1800
X-rays	1.5×10^{-6} to 7.0×10^{-10}	Roentgen; 1895
γ-rays	5.0×10^{-9} to 1.0×10^{-12}	Becquerel; 1896

FIGURE 2.

The Electromagnetic Theory: In the year 1853 Professor William Thomson (later Lord Kelvin), whose name is associated with the deep sea sounding machine and the laying of the first transatlantic cable, predicted that if an electrically charged condenser is discharged through an inductance the flow of current will not be undirectional; but will oscillate back and forth in the manner of a pendulum, which has been momentarily displaced from its

position of equilibrium. This prediction was verified four years later by Fedderson, who placed a spark gap in such a circuit and by means of a rotating mirror observed that upon discharge of the condenser a series of many sparks occurred in rapid succession.

These facts commanded the interest of Clerk Maxwell, the great mathematician, physicist, and first director of the famous Cavendish Laboratory at Cambridge. Maxwell set out to make a complete study of electric fields, and in due time was led to believe that when an electric field is established across a dielectric (such as air in a spark gap) there occurs an almost instantaneous flow of electric current. This so called *displacement current* then induces a magnetic field and the magnetic field begets another current, each followed by the other in a kind of vicious cycle, in a manner analogous to that of the proverbial dog who wags his tail and whose tail wags him. This idea is in accord with the induced magnetic field discovered by Oersted in 1819, and the converse effect of induced currents first observed by Faraday in 1831. It was on these two well established foundations that Maxwell built his electromagnetic theory and predicted that these electric and magnetic displacements should be propagated through space or through a dielectric in an oscillatory manner as waves.

A most extraordinary feature of this prediction was the calculation which indicated that these waves would travel in air with a speed of 193,088 miles per second; for, as he pointed out in a letter to Michael Faraday,[3] this value differed by only thirty miles per second from Fizeau's experimental value of the velocity of light. This coincidence naturally suggested to Maxwell that light itself might be an electromagnetic wave effect.

Now, as a matter of fact, the two values quoted above were both in error by more than six thousand miles per second; but the inference drawn was nevertheless correct, for when Maxwell detected and rectified the error in his calculations, his value of the velocity of the waves was in excellent agreement with the subsequently more refined measurements of the velocity of light. Had only the admirals Jellicoe and Beatty, in the first phase of the Battle of Jutland, been equally as fortunate in having the

[3] "James Clerk Maxwell," by Prof. J. J. Thomson and others (Macmillan Co., 1931), p. 102.

errors of their computed positions both in the same direction, their better condition of contact might have so hastened the termination of World War hostilities as did Maxwell's coincidence hasten his perception of the electromagnetic nature of light.

By this time, 1864, Maxwell strongly suspected light to be an electromagnetic phenomenon. It yet remained to establish as fact that an oscillating circuit would actually generate electromagnetic waves; for it must be remembered that this master of rare intuition and artist in the manipulation of differential equations had only sketched these waves upon his canvas.

Death overtook Maxwell in his forty-ninth year before his theory had received experimental verification; indeed, outside of a very few young men at Cambridge, only the two great princes of experimental research, Rowland of Baltimore and Helmholtz of Berlin, gave much credence to the electromagnetic theory. In 1887 Helmholtz' favorite student, Heindrich Hertz, then at the age of thirty a professor of physics at the Polytechnic School at Karlsruhe, performed an experiment which in years to come was to be "heard 'round the world." He had set up an oscillating circuit at one end of the lecture hall, at the other a sheet of metal which acted as a mirror, and between these a copper wire bent to form a nearly closed circle; and upon exciting the former a tiny spark jumped the gap of the detector. Not only so, but the rays emitted by the oscillator were focussed by a concave reflector, refracted by a prism of paraffin, and polarized by a grid of parallel wires. Here indeed was verification of Maxwell's remarkable theory; and, in Hertz' own words, "It also appears that the hypothesis as to the nature of light which is connected with that theory now forces itself upon the mind with still stronger reason than heretofore."

How these Hertzian waves were developed in the able hands of Marconi, DeForest, Edison, and Armstrong, is a story well known to radio engineers. The post-war alliance between the U. S. Navy and the wireless amateurs in developing short-wave communication and solving the skip distance puzzle[4] is one of the romances of radio. And now this full grown giant of commercial enterprise saturates the air, nay even the rooms and walls of our very houses,

[4] Taylor and Hulburt, Physical Review, 27, 189 (1926).

with its ubiquitous waves; each one, however, a direct descendant of a few differential equations in the lecture notes of one of England's most distinguished professors.

Waves in Matter: In the foregoing discussion similarity has been drawn between various types of radiation and ordinary light. The successful explanation of all these phenomena on the common basis of Maxwell's theory marks one of the most colossal achievements of theoretical physics and is now referred to as the classical theory of radiation. Mention was made in passing, however, of a certain unsatisfactory status of the photoelectric effect which, at the time, was postponed for further consideration. We are left, then, with the task of disposing of this topic as thoroughly as limited space permits and of drawing attention to the rather recent wave aspect of matter.

In the photoelectric effect it is observed that light, incident upon the surface of a metal, causes electrons to be emitted therefrom with a kinetic energy which depends only on that quality of light which in the wave theory is called its wave-length. Obviously the wave theory could not explain this fact, for the energy in a wave decreases with its distance from its source even while its wave-length remains constant. If one assumes, however, that light consists of discrete bundles of radiant energy which are projected through space like bullets from a machine gun, it is not difficult to perceive how such a bundle, or "quantum" as it is called, may impart all its energy to an electron at a single blow. Such a theory, first proposed by Professor Max Planck of the University of Berlin, implies a corpuscular nature of light. Thus we find ourselves confronted with a twofold aspect of radiation: one, particle-like and the other wave-like. In other words, one may say that radiation sometimes appears discretely distributed and sometimes spread continuously throughout space.

Let us now turn for just a moment to two distinct concepts of mass and see wherein they suggest a wave-like nature of matter. Firstly, the Greek philosophers were from early times accustomed to the integral quality of material objects, such as a sheep, a kernel of wheat, a grain of sand, etc., and consequently suspected that all matter is ultimately composed of indivisible particles. The atomic nature of matter was experimentally realized when, in

1808, John Dalton, a British schoolmaster, chemist, and scholar, published his great work, "A New System of Chemical Philosophy." Secondly, one of the most important single conclusions of Einstein's theory of relativity is that the energy content of a mass at rest is not zero but is numerically equal, in ergs, to the mass in grams times 9×10^{20}, a very large number indeed. From this result, cosmogonists have interpreted the radiant energy of the stars to be derived from the mass within where the pressures and temperatures are enormous; and it is estimated that the sun sacrifices 250,000,000 tons of mass every minute to maintain the tremendous radiation from its surface.

Combining now the idea of the discrete nature of matter and its transformation into radiation, it is not surprising that radiant energy is sometimes atomic or corpuscular, as suggested by the photoelectric effect. And conversely, one would suspect by virtue of the undalatory aspect of radiation that matter, too, would occasionally display a wave-like nature. Exactly this was recently found to be the case by Messrs. Davisson and Germer,[5] of the Bell Telephone Laboratories. These skillful investigators found that *electrons,* reflected from the crystal lattice surface of a piece of nickel, spread out in a diffraction pattern exactly as light rays from a diffraction grating. Professor A. J. Dempster,[6] of the University of Chicago, has shown experimentally that *protons* behave in an exactly similar manner. Thus the two fundamental building blocks of all matter, electrons and protons, display at times the characteristics of waves.[7]

In a world of radiation and matter where each of these constituents plays the dual role of wave and particle, it is immediately obvious that the electromagetic theory tells only half the story. To enter upon the other half, the new quantum theory, would be not only a breach of faith with the chosen topic of this discussion, but also an unwarranted duplication of a subject so adequately covered by recent expositors in physical science.[8]

The principles of strategy for naval warfare so superbly set forth by Mahan have perhaps found less comprehensive applica-

[5] Davisson and Germer, Physical Review, 30, 705 (1927).

[6] Dempster, Physical Review, 35, 298, 1405 (1930).

[7] Whatever may be the true nature of the recently discovered neutron and positron, there is ample reason to believe that they too will display wave characteristics.

[8] "Time, Space, and Atoms," R. T. Cox (Williams and Wilkins Co., 1933).

tion since the advent of the military airplane, yet the opinion is ventured that they are still fundamentally sound in the conduct and disposition of modern fighting fleets in time of war. In the same spirit it may be said that although the electromagnetic theory has lost some degree of generality by the comparatively recent discovery of new phenomena, it nevertheless remains the masterful genealogical record of a great family of waves, commencing at one end with the wireless wave, so long that a single one may easily span the distance between Baltimore and Washington, and terminated at the other by gamma rays, whose tiny undulations spontaneously emitted from radioactive material are so minute that two hundred and fifty million of them would barely cover an inch. When one perceives these various types of waves, radio, light, X-rays, etc., to be as different in their outward appearance as Dr. Jekyll and Mr. Hyde, one is often quite as astounded as in reading Stevenson's extraordinary case to find that they are fundamentally one and the same.

A NEW IDEA IN PUMP VALVE GEARS.

By Lieutenant William H. Galbraith, U. S. N., Member.

Some years ago, a sweating machinist's mate of the Navy served a cruise in an ocean going tug. As he remembers it, the principal battle of this cruise appeared to occur between himself and a number of reciprocating simplex pumps, equipped with conventional types of valve gear and apparently blessed with the temperaments of as many government mules. If it wasn't one thing it was another. Standing into port, for instance, the main feed pump would suddenly decide to stop, and after feverishly applying all known forms of first aid, it would be found that a valve wrench applied forcibly to the end of the valve stem imparted sufficient stimulus to bring forth a few more hesitant strokes. Under the circumstances, there was no alternative but for the machinist's mate to stand watch on the pump, wrench in hand and murder in his heart, until the dock should be made. And as the liberty party shoved off, he would prepare to spend the night with his friend, the pump.

He patiently catalogued the various ailments discovered each time he opened up a pump valve gear. In order of frequency of occurrence they appeared about as follows:

(*a*) So much steam leaked by the main valve and auxiliary valve that the valve piston, and even the main pump piston, scarcely knew which end should be receiving steam, and which exhausting.

(*b*) The main slide valve stuck because it had not worn evenly with its flat seat, against which the steam pressure held it so securely.

(*c*) An "ear" had broken off of a slide valve casting, or a nut had backed off in the interior assembly.

Each time, as he made his repairs, he cursed every thing that even faintly resembled an automatic valve gear and confided to himself that he could surely design a better device to accomplish

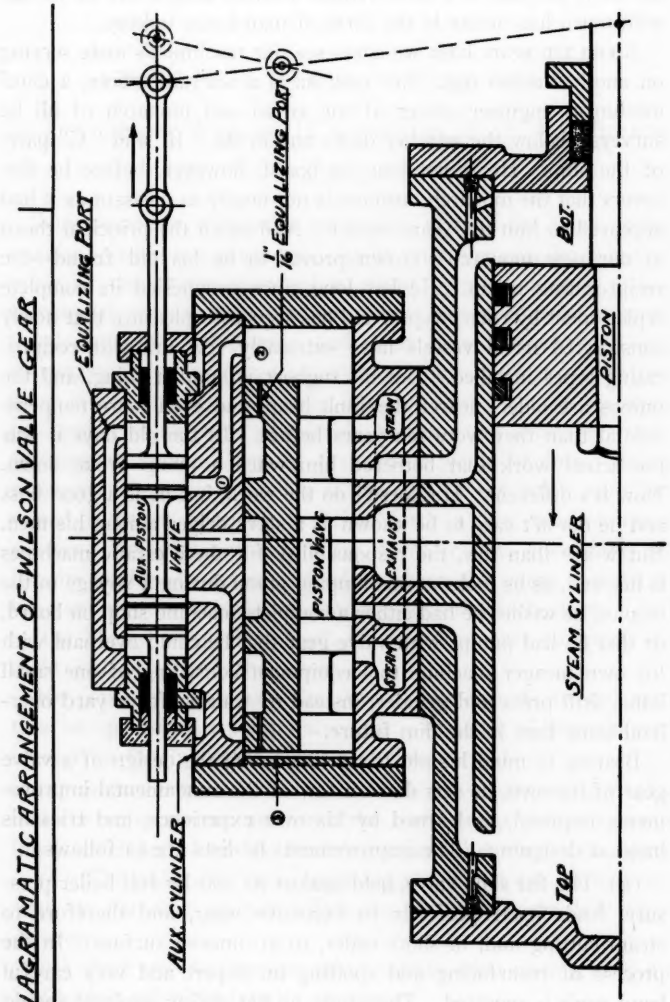
the same purpose in a more reliable manner and, above all things, with much less outlay in the form of man-hours upkeep.

About ten years later we again see our machinist's mate serving on another ocean tug. But now he is a warrant officer, a chief machinist, engineer officer of the vessel and monarch of all he surveys—below the weather decks and in the "B" and "C" parts of the ship. He is not long on board, however, before he discovers that the monarch business is not nearly as pleasant as it had appeared to him ten years earlier. And again the principal thorn in our new monarch's crown proves to be his old friend—the reciprocating pump. He has long since prophesied its complete replacement by rotary types, and has seen with pleasure that newly constructed naval vessels have extremely few units of reciprocating types installed. But his tugboat is another thing, and the pumps are older, nearer the junk heap, and even more temperamental than they were ten years before. In the old days it was the actual work that bothered him when a pump broke down. Now it's different. He doesn't do the work, but he is a good boss and he doesn't care to be known as a slave driver among his men. But worse than this, the responsibility for these erratic machines is his, and, as he looks at one limping pump during a voyage in the tropics, he wishes he had either a complete machine shop on board, or that he had pumps with valve gear that he could overhaul with his own meager machine tool equipment consisting of one small lathe, drill press and grinder, instead of waiting for a yard overhaul some time in the dim future.

Bearing in mind his old boast concerning the design of a valve gear of his own, he sets down a few of the fundamental improvements required, as learned by his own experience, and tries his hand at designing. The improvements he lists are as follows:

(*a*) The flat slide valve, held against its seat by full boiler pressure, lends itself perfectly to excessive wear, and therefore to steam cutting and, in short order, to an uneven surface. In the process of resurfacing and spotting in, expert and very careful hand work is required. Therefore, no flat sliding surfaces should be employed.

(*b*) Wear of valve and seat is caused by large unbalanced pressures. Therefore a piston valve of the fully balanced type must be used.



DIAGRAMATIC ARRANGEMENT OF WILSON VALVE GEAR

(*c*) In as much as the valve gear must be examined and overhauled much more often than the steam cylinder, the valve chest complete, including valve and seat, must be an entirely separate unit from the steam cylinder, capable of ready removal.

(*d*) Internal parts should be of one integral piece thereby obviating the chance of the backing off of a nut, bolt, stud or other separate part within the gear. This requires that the main valve and its piston be machined on the same billet and that there be no separate adjacent moving parts.

(*e*) The design must be such as to permit a thorough overhaul to be accomplished with the aid only of ordinary machinists' hand tools plus a small lathe.

(*f*) For economical considerations, the gear must be adaptable to the steam cylinders of simplex pumps now installed in naval vessels.

The accompanying figure shows, diagrammatically, the resulting design. Referring to this figure the gear operates in the following manner:

When the pump piston reaches the top of its stroke the auxiliary piston valve is moved down, opening port (1) to the annular exhaust space about the center of the auxiliary and main piston valves, thus releasing pressure in space (2) and permitting the unbalanced pressure in space (3) to force the main valve piston down. This piston cushions itself upon closing port (1). The initial condition of steam balance is immediately reestablished by means of the small 1/16-inch equalizing port shown. The action of the main valve and the corresponding pump piston movement is obvious. The motions described above are, of course, repeated at the opposite end of the stroke.

The force actuating the main valve piston is obtained by virtue of the difference in the rate of flow of steam through port (1), which is 1/4 inch in diameter, and the 1/16-inch equalizing port drilled through the outside collars of the main piston valve. It will be noted that, except when the main valve is actually in motion, it is in a state of complete balance, both axially and circumferentially. The small equalizing port, connecting the outer ends of the auxiliary valve cylinder, is necessary to permit easy movement of this valve. In this connection, it will be seen that

the valve actuating rod need be packed but lightly for it is called upon to seal only against the auxiliary exhaust pressure.

While the figure has been made diagrammatical for the sake of clearness, it is, nevertheless, quite faithful as to the principal details of construction. It will be seen that the entire chest may be removed, chucked up in a small lathe and rebored if it should show any sign of uneven wear. The diameters of both the main and auxiliary valve chests and pistons are the same throughout to facilitate both this operation and that of turning new pistons from any billet of sufficient diameter and length.

Thus we have what is becoming known as the Wilson Valve Gear. It was experimentally built and applied to a standard 7 inches $\times$ 7 inches $\times$ 12 inches fire and bilge pump and was operated almost continually for a period of about 4000 hours in an ocean going tug. It gave perfect results and upon measuring its parts, no appreciable wear could be detected following this service. While under test it was found that the auxiliary valve stem could be readily moved by hand, thus arresting the pump and reversing it in mid stroke.

The gear is applicable to any reciprocating steam pump employing automatic valve gear. The valve chest simply bolts into place over a 1/32-inch copper gasket without alteration to the steam cylinder of the pump.

With the increasing use of superheat in naval vessels, a further advantage of the Wilson Valve Gear appears in its ready adaptability to the insertion of a hardened steel liner in the main valve cylinder, the main valve then to be turned from a steel forging. The auxiliary valve materials, operating only in wet exhaust steam, may remain the same.

Further tests and improvement of this valve gear are now under way and it is to be hoped that its use may speedily rejuvenate the many wheezy pumps now in service and thus contribute to the general serenity of the engineer personnel of the Navy.

THE CALIBRATION AND OPERATION OF TANK GAGES.

By COMMANDER H. F. D. DAVIS, U. S. N., MEMBER.

Some questions which are puzzling until carefully gone into may arise in connection with the calibration and operation of tank gages. It is therefore desirable to make available, for the ready reference of those who may be faced with the problem of calibrating tanks and gages and those who desire to use them properly, an explanation of these gages and of the methods of calibration and use.

Calibration of a tank. The tank with which the tank gage is to be used must first be "calibrated." If the tank is of regular shape the process may simply consist in computing the volume in cubic feet or gallons corresponding to each height at which liquid may stand. If the tank is irregular it may be necessary to put in measured quantities of a liquid and measure the height for each

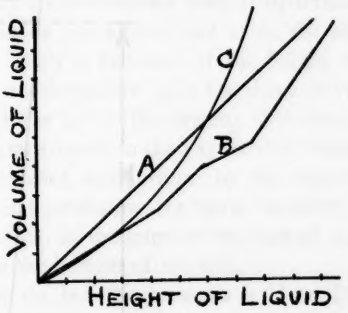


Fig. 1. Tank Calibration Curves.

known quantity added, the amounts added in each step depending on the accuracy desired. This calibration may then be put in the form given in Figure 1, in which A, B and C represent some of

the possible variations which the calibration may take; or a table may be made for the different values of

$$V = f(H) \qquad (1)$$

Then from the table, the graphs, or the equation, the volume of any liquid in the tank may be obtained if the height, H, is known. The tank gage, often a mercury gage, but in which other liquids may be used, is the simple device provided primarily to determine H by reading the gage, thereby obviating the necessity of using a sounding rod, tape, or other method of measuring the height. However, in order to use the reading of the gage correctly, a full knowledge must be had of the relation of such reading to the height and to the other variables in the problem, including of course the volume which is ultimately to be determined.

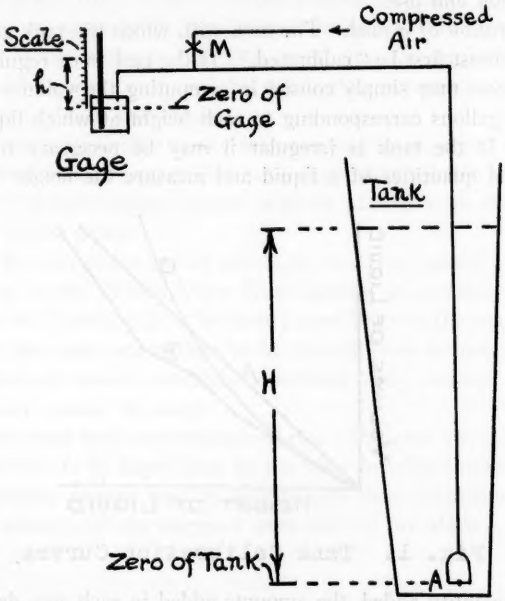


Fig. 2. Diagrammatic Arrangement of Tank and Gage.

Refer now to Figure 2 which illustrates a tank in which liquid is standing to height H and occupying volume V with gage connected to the tank by the piping as indicated.

Operation of the Gage. With no pressure in the piping connecting the gage to the bell in the tank, the column of liquid in the gage will be at zero level in the reservoir.

The valve M is opened and by means of a pump or other source of compressed air, pressure is put on the line connecting the gage and bell until the air displaces all the liquid in the pipe and escapes from the bell at the bottom of the tank. This can be determined by the fact that the liquid will cease to rise in the gage.

Caution. It is obvious that the air pressure must not be put on too rapidly otherwise the inertia of the liquid in the tank may cause the pressure in the pipe and in the gage to rise momentarily high enough to blow the liquid out of the gage. Some means may be provided in design to reduce to a considerable extent the possibility of this casualty.

The air pressure having been put on gradually and the liquid in the gage having ceased to rise, valve M is turned off. If there are no leaks in the system the liquid in the gage will remain at the same height.

In accordance with the well-known laws of hydraulics the intensity of pressure, pounds per square foot at A, the bottom of the bell in the tank, is solely a function of the height of the liquid and its density, and not dependent upon the shape or volume of the tank. By means of the air in the system, this pressure, pounds per square foot, is transferred to the gage and is balanced therein, when the system reaches equilibrium, by the column of liquid rising to height h and producing the same intensity of pressure, pounds per square foot, in the gage at the foot of the column as exists in the tank at the bottom of the bell.

At the bottom of the bell the pressure is $H \times D$ pounds per square foot, and at the foot of the gage column the pressure is $h \times d$ pounds per square foot, D being the density, pounds per cubic foot, of the liquid in the tank and d the density, pounds per cubic foot, of liquid in the gage. It therefore follows that

$$h\,d = H\,D \qquad (2)$$

$$\text{and } h = H\,D \div d \qquad (3)$$

If S is the specific gravity of the liquid in the tank and s the specific gravity of the liquid in the gage, equation (3) reduces to

$$h = H\ S \div s \qquad (4)$$

The foregoing relations are general and true for any liquids that may be put in the tank or used in the gage.

Calibration of the Gage. The gage may of course have a scale in feet or inches alongside the column which will be called the "main" scale. The zero of this "main" scale is the level of the liquid in the reservoir with no pressure in the system. This scale can be made a part of the gage and essentially bears no relation to the tank with which the gage is to be used.

Suppose, however, that it is desired to have another "auxiliary" gage scale indicating cubic feet, gallons, or tons of liquid of a given density. Since, as shown above in establishing equations (2), (3) and (4), the height of the liquid in the gage depends on the density of the liquid in the tank as well as on the height, these auxiliary scales can only be correct for a liquid of specified density (or specific gravity).

A convenient standard for the marking of the auxiliary scale is to calibrate for water at 60 degrees F., weighing 62.3 pounds per cubic foot, and to call this liquid of unity specific gravity.

Using this standard, the auxiliary scales may be made up by the following procedure: (A mercury gage will be assumed).

(a) For height H (feet) of liquid (unit specific gravity) in the tank, the height of mercury column h_w (feet) will be

$$h_w = H\ S_w \div s = H \div 13.6 \qquad (5)$$

$S_w = 1$ and $s = 13.6$ being the specific gravities of water and mercury.

(*b*) Corresponding to H (feet) is the volume V (cubic feet), known from the calibration of the tank. If the auxiliary scale is to be in cubic feet, the auxiliary scale should be marked V cubic feet opposite h_w (feet) on the foot scale. This process is repeated for all values of H and V.

(*c*) If an auxiliary scale in gallons is wanted,

$$\frac{V \times 1728}{231} = X \text{ should be marked opposite } h_w$$

(*d*) If an auxiliary scale in long tons of water, 62.3 pounds per cubic foot, is desired

$$\frac{V \times 62.3}{2240} = Y \text{ should be marked opposite } h_w$$

Use of Scales for Any Other Liquid. For a liquid of specific gravity S in the tank to a height H, as shown above, the gage will read

$$h = HS \div s \qquad (4)$$

The auxiliary scales having been marked for water of standard density, they will not in the general case, indicate correctly for liquid of another density. To get the correct volume it is first necessary to determine h_w, the height at which the mercury would stand if H feet of standard water were in the tank instead of H feet of another liquid.

From equation (4) we have

$$H = hs \div S \qquad (4)$$

From equation (5)

$$H = h_w s \qquad (5)$$

Therefore $h_w = h \div S$ (6)

Then the procedure to obtain the correct volume is obviously

(*a*) Read the gage in feet (or inches) obtaining h

(*b*) Determine the specific gravity of the liquid, S

(*c*) Divide h by S to obtain h_w

(*d*) Opposite h_w (on the main scale) on the auxiliary scale of cubic feet, gallons, or tons, read the desired answer.

(*e*) If the liquid in the tank is not at 60 degrees F. and the volume at 60 degrees F. is desired, reduce S to S_{60} (see Tables 30-39, Chapter 39, Manual of Engineering Instructions) and proceed as in (*c*) and (*d*).

For example, if V = KH, K being a constant, with standard water in the tank to height H, the gage scale reads h_w and indicates proper volume V = K H on the auxiliary scale. When liquid of 0.9 specific gravity is put in the tank to height H, the

main scale reads $h = 0.9\ h_w$, indicates only 0.9 H depth of liquid and on the auxiliary scale reads $V = 0.9\ K\ H$. The correct volume can be obtained by dividing h by 0.9 to get h_w and then reading the auxiliary scale in accordance with the procedure given above; or obviously, for this case, the reading on the auxiliary scale opposite h, i.e., 0.9 K H, if divided by 0.9 will give the correct volume $V = K\ H$. Thus, when the volume of the tank is a straight line function of the height of liquid in the tank, a single scale will be sufficient. *By carrying through the correct procedure, then the alternative and making a comparison of results, a test* of the propriety of the alternative method may be quickly made for any tank and its gage. Such tests may be made by simply choosing arbitrarily a specific gravity and working out theoretical cases for various depths of liquid in the tank.

If, however, $V = f\ (H)$ is not a straight line relation, the alternative procedure is not correct. Thus, suppose $V = K\ H^3$. When liquid of 0.9 specific gravity is put in the tank to height H, the main scale of the gage reads $0.9\ h_w$, indicates only 0.9 H height of liquid in the tank, and therefore only volume $V = K\ (0.9\ H)^3 = 0.729\ K\ H^3$ on the auxiliary scale. This is far from the correct volume, $V = K\ H^3$.

Density or Specific Gravity Determinations. The theoretical basis for the determination of the density or specific gravity of the liquid in a tank by means of tank gages is of interest.

In addition to the ordinary gage which extends to the bottom of the tank, another may be put into the tank at about one-half total depth, the zeros of the two bells being at a fixed vertical height, P feet, apart. Suppose liquid is not put in the tank.

If h_1 is the reading of the first gage indicating a height H_1 of liquid density D, from equation (4) it follows that

$$h_1 = H_1 \frac{D}{d} \text{ or } h_1 = H_1 \frac{S}{s}$$

Likewise, if h_2 is the reading of the second gage indicating a height H_2 of liquid density D, it follows that

$$h_2 = H_2 \frac{D}{d} \text{ or } h_2 = H_2 \frac{S}{s}$$

subtracting the second equation from the first

$$h_1 - h_2 = \frac{D}{d}(H_1 - H_2) \text{ or } h_1 - h_2 = \frac{S}{s}(H_1 - h_2)$$

from which

$$D = \frac{(h_1 - h_2)\,d}{(H_1 - H_2)} = \frac{(h_1 - h_2)\,d}{P} \text{ or } S = \frac{(h_1 - h_2)\,s}{P}$$

Since all the quantities of the right-hand side of the equations are known when readings h_1 and h_2 are taken, the values of D or S may then be computed.

Errors. As with any instrument certain errors must be guarded against in tank gages.

When the zero of the gage differs in level from the zero of the tank (bottom of the bell) (in ship installations the gage is usually considerably higher) the pressure at the lower of the two levels will be greater by the amount of the pressure due to the head of air. Since, however, air has only about one ten-thousandth the density of mercury, a very large difference in levels of the zeros may exist without producing errors in the readings to which any attention need be paid.

The zero of the mercury gage must be carefully marked and the mercury kept adjusted to that point. The level in the reservoir falls as the column goes up, and, if the size of the column is large as compared with the size of the reservoir, an error is introduced, unless compensated for by movement of the scale or special marking thereof. Suppose, for example the mercury reservoir has a cross sectional area of one square inch and the tube a cross sectional area of one-hundredth of a square inch. When the mercury rises one inch the volume in the column is a hundredth of a cubic inch and the surface in the reservoir therefore falls one-hundredth of an inch. If the main scale was calibrated to read one inch from the original surface an error of one per cent would be introduced. The scale may therefore be made to indicate one inch at 0.99 inch above the "no pressure" zero of the gage; the two-inch indication may be at 1.98 inch above the original zero and so on. The auxiliary scales in cubic feet, gallons, or tons of tank contents must likewise be adjusted to provide for the changing zero of the mercury column.

A leak in the system obviously allows the mercury to fall after the equilibrium has been established, rapidly or slowly, depending on the size of the leak. Check observations after an interval of time will of course indicate such leaks.

The piping and gage should be of sufficient size to minimize the effects of capillarity, cohesion, etc.

As compressed air usually contains some moisture, care must be taken to guard against the entry and condensation of moisture in the lines and gage and to provide for the removal of any that accumulates.

The specific gravities of the liquids should be determined with a precision consistent with the results desired.

On board ship changes in trim of the ship will affect the accuracy of the readings to a greater or less amount depending on the position of the gage in the tank and, of course, on the amount of change of trim from even keel.

Checking of gages. In order to check the readings of gages by soundings or measurement of ullage it is necessary that the distances of the zero of the tank to the zero of the sounding tube, and to the zero of the point from which the ullage is measured should be carefully fixed. With this caution in mind and with the ship on even keel, or with allowance made for any variation from even keel, the procedure for checking of gages is obvious.

SUMMARY.

The foregoing may be summarized as follows:

(*a*) The volume of liquid in the tank, V, is a function of the height, H,

$$\text{i.e., } V = f(H) \qquad (1)$$

The relations expressed by equation (1) must be obtained by "*calibration*" of each tank to which a gage is to be connected; this is fundamental and essential information.

(*b*) To obtain the height the liquid stands in the tank, H, by means of a tank gage, use the relations

$$Hd = HD \quad \text{or} \quad hs = HS$$

$$H = hd \div D \quad \text{or} \quad H = hs \div S$$

$$h = HD \div d \qquad h = HS \div s$$

It should be noted that these relations depend on the densities of the liquids in tank and gage.

(*c*) To obtain the height, h_w at which the gage column would stand, if liquid of unity specific gravity were in the tank to height, H, use equation

$$h_w = h \div S$$

(*d*) A main scale of feet (or inches) at the gage, simply allows the determination of H by means of the above equations; to obtain the volume V or the corresponding weight requires a calibration table, or graph of $V = f(H)$.

(*e*) Auxiliary scales of volume (cubic feet or gallons) or weight (tons) can be placed alongside the feet (inches) scale of the gage, but will be correct only for liquid of a given density. Such scales in effect simply place the tank calibration data in a position for convenient use.

(*f*) When $V = f(H)$ is not a straight line, to obtain the correct volume (or weight) from an auxiliary scale, first compute h_w, then opposite h_w on the main scale read the correct volume on the auxiliary scale.

THE TREND IN NAVAL ENGINEERING.

Alternating-Current Motors and Generators.

Part 3.—Generators.

By C. Huey, Civil Member.*

In dealing with the subject of alternating-current generators for shipboard use in Naval Service, the number of *basic* types to be considered is very limited. In fact, the number available in the commercial field is limited to two, known as, (*a*) the *induction* or asynchronous generator, and (*b*) the alternator or *synchronous* generator.

In describing the operating principles of the two basic types of alternating-current generators, it is convenient to refer to the writer's previous article on alternating-current *motors*. The terms *induction* and *synchronous*, as applied to *motors*, indicate definite operating principles. These motors, when driven by mechanical power, function as A.C. generators. Therefore, structurally, induction generators and synchronous generators, are identical with the corresponding motors. *Synchronous* generators may be subdivided in four types. Collectively, the various forms of alternating-current generators are listed as follows, and each will be described in turn.

(*a*) Induction Generator (Asynchronous). See Plate A.

(*b*) Inductor Generator (Synchronous). See Plate B.

(*c*) Revolving-Armature Generator (Synchronous). See Plate C.

(*d*) Inverted Rotary Converter (Synchronous). See Plate D.

(*e*) Revolving-Field Generator (Synchronous)

(1) Salient-Pole Type. See Plate E.

(2) Embedded-Coil Type. See Plate F.

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THE INDUCTION GENERATOR.

In the case of the *induction motor,* if the armature or *primary* winding is connected to A.C. power lines, and if the *secondary* winding or rotor is driven above synchronous speed by an engine or other mechanical power, then an opposing A.C. current, at line frequency, will be generated in its *primary* winding, and will feed back into the lines. An induction motor used in this manner becomes an induction *generator.* The exciting current required for the current generation is supplied by the A.C. lines. The induction generator has no properties of controlling the frequency of the power lines which it feeds. Therefore, it is properly termed an *asynchronous* generator, meaning, lacking in synchronism with some associated apparatus. The effect of increasing its speed above the plant frequency synchronous speed, is to feed more current into the system. If its mechanical driving power fails, the generator will continue to run below synchronous speed as an induction *motor,* and load current will be drawn from the line sufficient to balance the friction requirements of the motor and its prime mover. Figure 1 illustrates the *circuit* relations of an induction generator.

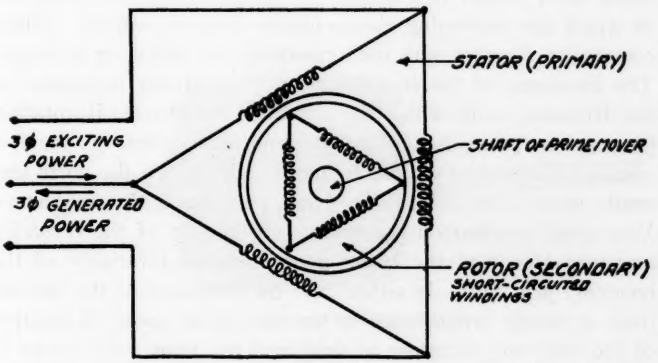


FIGURE 1.—INDUCTION A.C. GENERATOR, THREE-PHASE, CIRCUIT RELATION OF ROTOR AND STATOR.

APPLICATIONS OF INDUCTION GENERATORS.

The peculiar features of induction generators militate against their use as primary sources of alternating-current, for the reasons that they depend on A.C. excitation established by other sources, and the power that is generated by them must be fed into the circuits of the exciting source and at the frequency of the exciting source. Induction generators are used to some extent in power plants to share the minority part of the main *synchronous-generated* load. In such installations, induction generators serve to greatly reduce the effect of heavy overloads caused by line short-circuits and other severe disturbances. Also due to the fact that varying the speed of induction generators has no effect on the frequency when paralleled with synchronous generators, they have been found practical and economical when driven by ungoverned steam turbines from exhaust steam systems in commercial applications. In such installations, the induction generator feeds the line up to the available capacity of the steam power.

THE SYNCHRONOUS GENERATOR.

A synchronous generator usually consists of a system of alternately north and south magnetic poles comprising the *field,* which moves with respect to a system of suitably connected conductors in which the alternating electro-motive force is *induced.* These conductors, together with their mounting, are called the *armature.* The frequency of the alternating e.m.f. is directly dependent on the frequency with which the armature conductors, if rotating, pass across the fields of the north and south poles, or, the frequency is dependent on the frequency with which the north and south poles of the field, if revolving, pass across the conductors. Very small synchronous generators are usually of the *revolving armature* type, and the larger generators are invariably of the *revolving field type.* In either case, the frequency of the alternations is strictly proportional to the mechanical speed of rotation of the revolving armature or field, and the term *synchronous* is associated with speed or R.P.M. of the power shaft. The actual frequency in cycles per second is the product of the number of *pairs* of poles (north and south) and the revolutions per second of the rotating element of the generator.

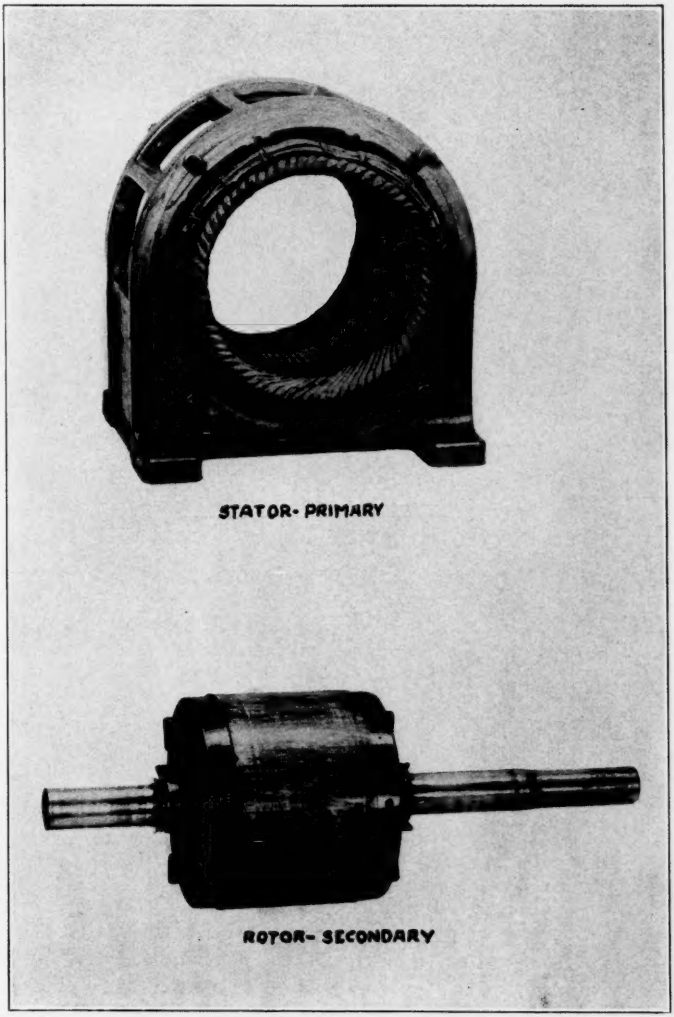


PLATE A—THE INDUCTION GENERATOR.

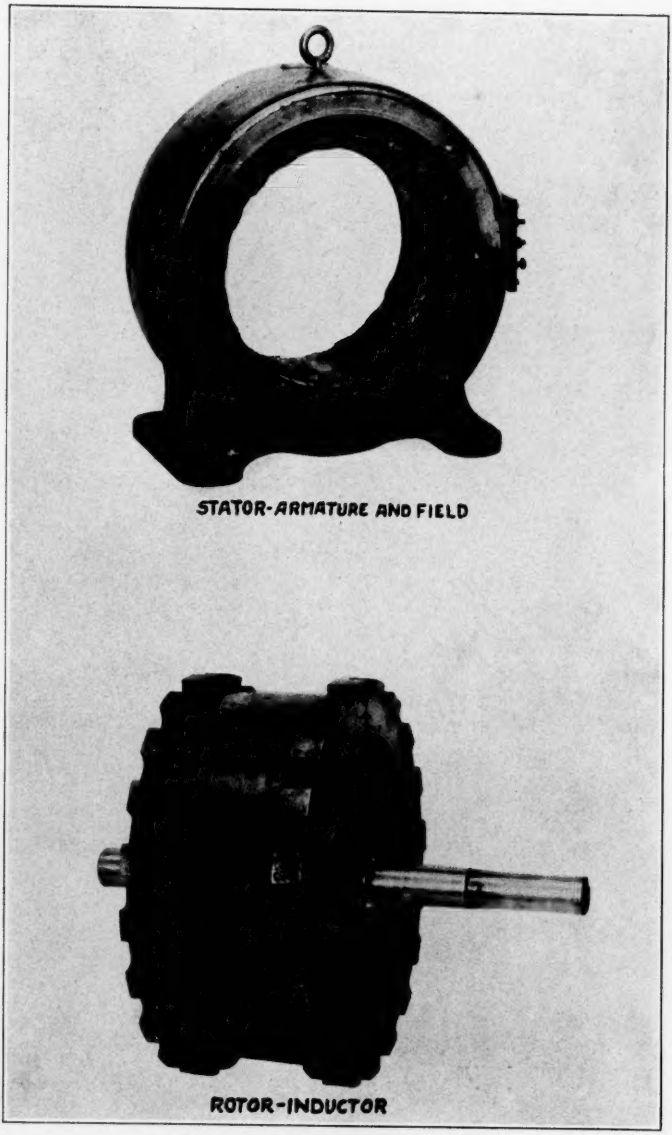


PLATE B—THE INDUCTOR GENERATOR.

THE SYNCHRONOUS GENERATOR, INDUCTOR TYPE.

One type of synchronous generator which has been used in small sizes for radio power and other special application, is known as the *inductor* generator. This type of generator does not conform to the foregoing description of the usual synchronous generator, as both the field and the armature windings are *stationary*. In this machine the revolving element consists of masses of iron or inductors which are continuously magnetized across an air-gap, by a direct-current stationary annular field coil. As the pole tips of the inductor pass across the stationary armature coils, the magnetic flux through the coils varies in intensity from minimum to maximum. This in turn induces alternating e.m.f. in the armature coils. These machines may be either single or polyphase, and require direct-current excitation. The basic scheme which causes alternating voltage to be induced in the armature coils by the rotation of the inductor rotor is shown in Figure 2, illustrating a single phase, double circuit generator.

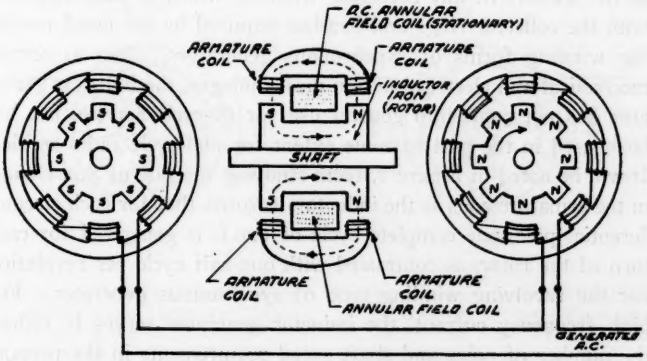


FIGURE 2.—THE SYNCHRONOUS INDUCTOR A.C. GENERATOR.

Referring to the left-hand side view of Figure 2, when the south poles of the inductor travel from one armature pole to the adjacent armature pole, the magnetism through the armature coils changes from maximum in one direction to a minimum in the same direction. Thus, alternating electro-motive-force is generated in accordance with an electro-magnetic law which may be stated as

follows: Electro-motive-force, of a definite polarity, is generated in a conductor or coil when located at right angles to a magnetic field of *increasing* or *decreasing* intensity; the e.m.f. so generated is proportional to the *rate of change* of the magnetic flux. Therefore, referring again to Figure 2, the opposite side of the inductor with the north poles, is likewise inducing alternating e.m.f. in the duplicate armature coils under the same electro-magnetic law of *varying* magnetic intensity. The two armature circuits may be used separately for single voltage, or may be externally connected in series for double voltage service. The frequency of the alternations is directly dependent on the speed of the inductor. The inductor generator, therefore, may be properly classified as a *synchronous* generator, for the reason that the frequency of alternations at all times is synchronous with the speed of mechanical rotation.

APPLICATION OF THE INDUCTOR GENERATOR.

The inductor generator has several theoretical advantages, such as the absence of any revolving winding, which in turn dispenses with the collector rings and brushes required by the usual revolving winding forms of synchronous generators. Due to certain mechanical and electrical design disadvantages, the inductor generator has not come into general use for 60-cycle service, but has been used in the past to some extent for 500-cycle radio service. It will be noted in Figure 2, from studying the rate of flux change in the armature coils as the inductors revolve, that for each circumferential pole, one complete cycle of e.m.f. is generated for each turn of the rotor, as contrasted with one-half cycle per revolution for the revolving winding type of synchronous generator. For high frequency current, the inductor generator serves to reduce the number of poles and shaft speed requirements in the proportion of 2 to 1 compared with the revolving winding-type synchronous generators.

THE SYNCHRONOUS GENERATOR, REVOLVING ARMATURE TYPE.

The *revolving-armature* type generator, is the one frequently used for small capacity generation of alternating-current, either single phase of polyphase. In this form, the principle of opera-

tion is practically the same as the usual *direct-current* generator. In the case of the *direct-current* generator, *alternating* e.m.f. is induced in the revolving armature as its coils cut through the magnetic flux of the alternate north and south poles of the surrounding D.C. excited field system. By employing a commutator, to which the individual armature coils are connected, and by proper location of brushes on the commutator, the current generated is conducted to the external circuit before the reversal of the e.m.f. (see Figure 3). If the same armature were to be connected in a different manner, so that taps from the winding are connected to slip rings or collector rings as in Figure 4, then a single phase alternating e.m.f. will be available across the rings which in turn may be carried to the external circuit by means of stationary brushes. The frequency in cycles per second of alternating-current so generated, is directly dependent on the number of *pairs* of D.C. field poles (one north and one south) and the revolutions per second of the armature.

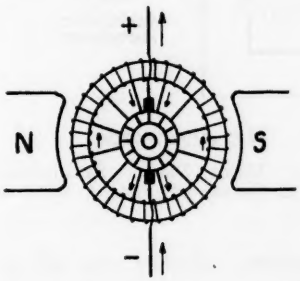


FIGURE 3.—DIRECT-CURRENT GENERATOR WITH REVOLVING ARMATURE, COMMUTATOR AND BRUSHES.

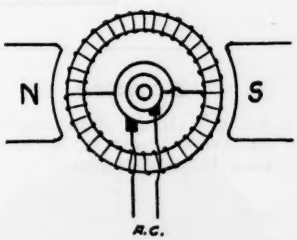


FIGURE 4.—ALTERNATING-CURRENT GENERATOR WITH REVOLVING ARMATURE, SLIP-RINGS AND BRUSHES.

APPLICATION OF THE REVOLVING ARMATURE TYPE.

In the *revolving-armature* type synchronous generator, the necessity for conducting the generated current from the revolving winding, by means of slip-rings and brushes, limits its use to generators of very small current capacity. In fact, even in very small capacity generators of this general form, it is almost the

invariable rule to revolve the *field* winding in preference to the armature. In some applications, however, practical considerations require the use of the *revolving-armature* type. An example of this is in direct-current *motor-driven,* A.C. generators, where the voltage regulation of the generator is greatly improved by circulating the D.C. motor main-current through series windings of the stationary field system of the generator, as shown in Figure 5. In this scheme, the D.C. shunt field of the A.C. generator is augmented by the motor D.C. current, in proportion to the A.C. generated power, in effect, the same as a compound wound D.C. generator.

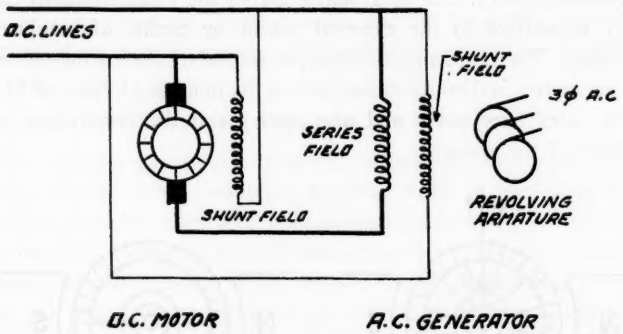


FIGURE 5.—CIRCUIT ARRANGEMENT OF MOTOR DRIVEN SYNCHRONOUS REVOLVING-ARMATURE, 3-PHASE A.C. GENERATOR, FIELD COMPOUNDED FROM MOTOR CURRENT.

Another application of the *revolving armature* type of synchronous generator, is known as an *inverted rotary converter.* Strictly speaking it is not a *generator,* but from a machine standpoint, it belongs properly in this classification. This type of machine *converts* direct-current into alternating current. The conversion is accomplished by the use of a D.C. shunt motor having the usual revolving commutator, wound armature, and speed control by means of a shunt-field rheostat. Alternating-current, at a frequency depending on the speed and number of poles, is obtained by taps from the same armature winding. These taps are connected to slip-rings from points on the armature

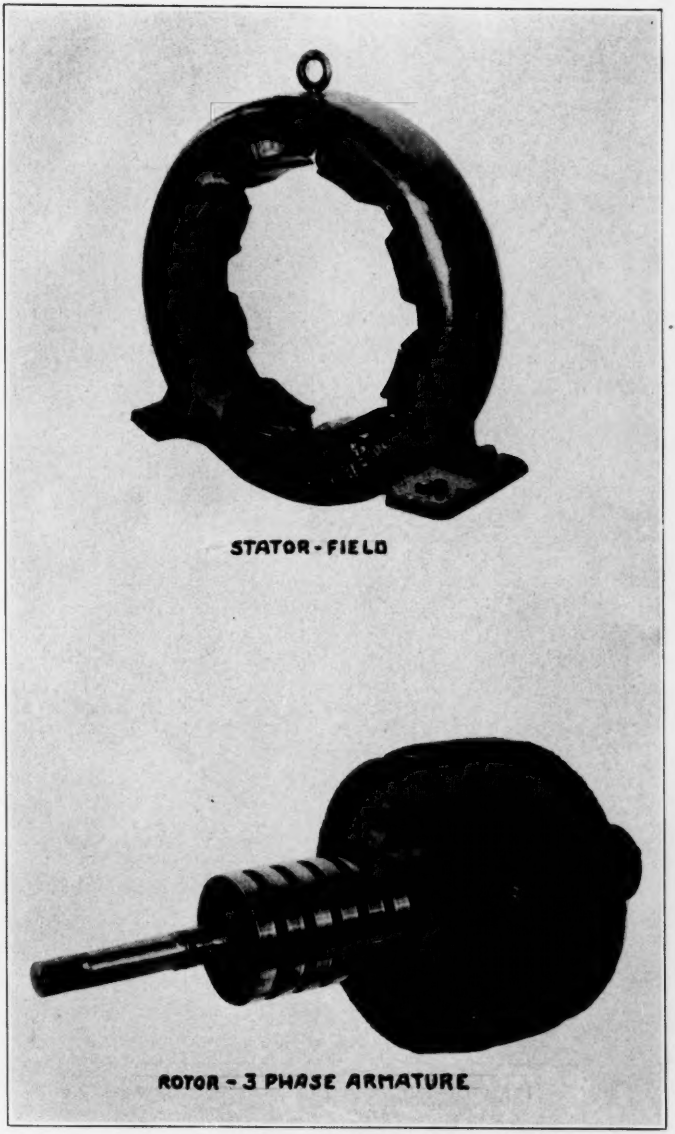


PLATE C—THE REVOLVING-ARMATURE GENERATOR.

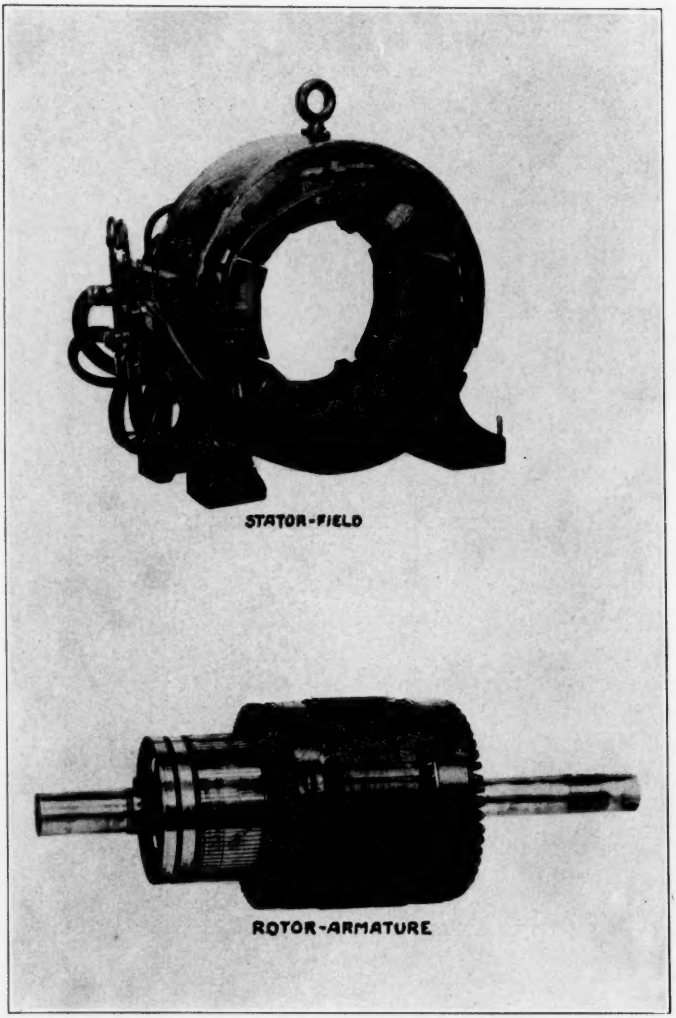


PLATE D—THE INVERTED ROTARY CONVERTER.

winding, 180 electrical degrees apart for single-phase service, and 120 electrical degrees apart for three-phase service, as shown in Figure 6.

In the conversion of D.C. to A.C. by the rotary converter method, the D.C. applied volts equals the converted maximum or peak A.C. volts. In terms of *effective* A.C. volts, the conversion factor for single-phase, is .707 times the D.C. volts. For con-

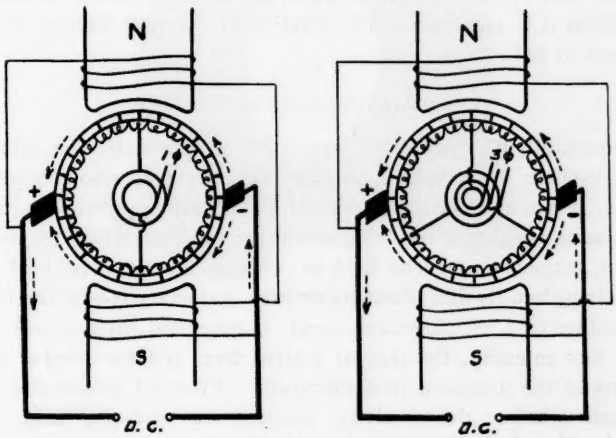


FIGURE 6.—D.C. TO A.C. ROTARY CONVERTER, SHOWING D.C. COMMUTATOR, ARMATURE WINDING, A.C. SLIP-RINGS, AND TWO-POLE, D.C. EXCITATION.

version to three-phase A.C., the vector relation reduces the factor to .612 times the D.C. volts. The *inverted* rotary converter may be operated in the *reverse* manner to convert A.C. to D.C. The machine operated this way is termed a *rotary converter*. In this operation the motor is run as a synchronous A.C. motor excited from its own D.C. commutator or from a separate D.C. source. The conversion factor in this case is D.C. volts = 1.41 A.C. volts for single phase, and 1.63 A.C. volts for three-phase. The voltage conversion factors given above, are definitely fixed values for sine-wave conversion. These values cannot be changed by variation of the D.C. field excitation of either the converter or inverted

converter system. In the former, variation in the field strength serves only to change the power-factor of the synchronous motor, and in the latter, it serves to change only the speed of the D.C. motor and the frequency of the converted A.C. In actual practice, the converted voltage will always be less than the conversion factor, due to the electrical losses when current is transmitted to an external load. Single-phase inverted rotary converters have been used to some extent in Naval Service as a source of independent A.C. current for the operation of electrical Salinity Indicators on D.C. ships.

THE REVOLVING-FIELD GENERATOR.

Practically all synchronous generators of appreciable size, either single-phase or polyphase, are the D.C. excited, *revolving-field* type. From an electrical standpoint, the operating principle is just the same whether a revolving conductor coil cuts across the flux of a stationary magnetic field or whether the magnetic field is revolving and its flux is cutting across a stationary conductor coil. In either case, an alternating e.m.f. is generated proportional to the flux intensity, the rate of cutting flux, and the number of turns in the armature conductor coils. Practical considerations dictate whether the revolving armature or revolving field is preferable. It is obvious for the larger sizes, where large generated current is to be transmitted, where high generated voltage is to be insulated, and where slip rings and brushes must be used in either system, that it is more practicable to arrange for the stationary armature, and use the slip rings and brushes for the relatively small power requirements of the D.C. excited revolving field. The revolving-field type exists in two forms, (*a*) salient poles and, (*b*) embedded coils.

THE SALIENT-POLE REVOLVING FIELD.

A salient pole means a *projecting* or *conspicuous* pole. A revolving field of this form, is constructed of pole structures bolted or keyed to a shaft spider with the D.C. current carrying circuit insulated and wound around each projecting pole core. The field windings are usually all connected in series and the ends of the

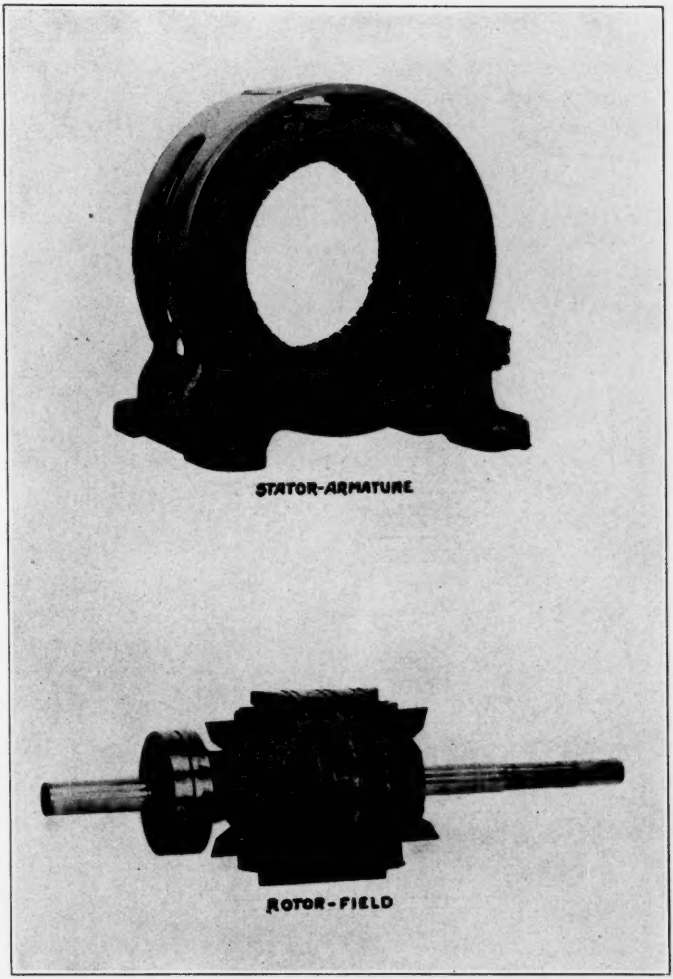


PLATE E—THE SALIENT-POLE REVOLVING-FIELD GENERATOR.

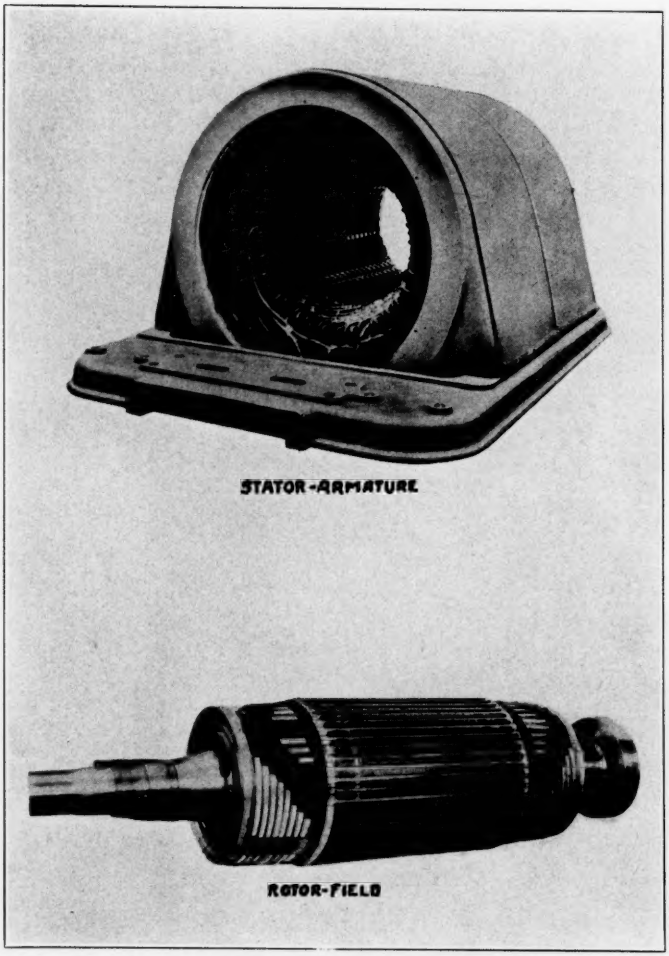


PLATE F—THE EMBEDDED REVOLVING-FIELD GENERATOR.

circuit are connected to two slip-rings insulated from and mounted on the rotor shaft as shown in Figure 7(a). The salient pole type of construction is used chiefly for low speed designs, to be expected with prime movers such as reciprocating engines, geared turbines, water wheels, or electric motor drives. The volumetric space occupied by the salient pole construction, and the practicability of using numerous salient poles to obtain commercial frequencies at low shaft speeds, serves to permit the use of a large diameter stator frame and armature assembly. This assembly provides ample space for insulation and ventilation, so that high voltages or high currents may be employed to advantage.

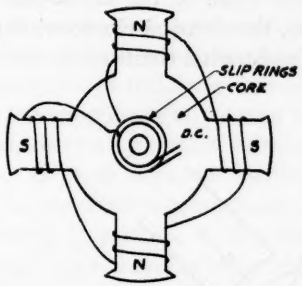


FIGURE 7(A).—SALIENT-POLE TYPE OF REVOLVING FIELD, 4 POLES.

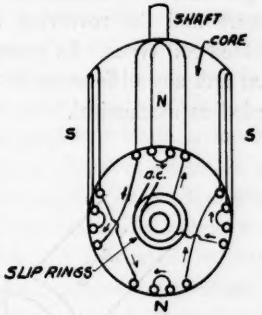


FIGURE 7(B).—EMBEDDED-COIL TYPE OF REVOLVING FIELD, 4 POLES.

THE EMBEDDED COIL REVOLVING FIELD.

The advent of the high-speed, *direct-connected* steam turbine for the prime mover, brought with it the design limits of the salient pole construction. For the higher speeds, the revolving field structure consists of a relatively small diameter cylindrical rotor with the windings *embedded* in slots, so arranged as to form usually either two or four magnetic poles as shown in Figure 7(b). At high shaft speeds, this construction obtains better balance, less noise, less windage and is stronger mechanically. Naturally, with the small diameter rotor, its *length* is greatly extended in order to obtain the required field strength, and the surrounding stationary armature is required to have deep slots and forced ventilation, to

provide for the current and voltage requirements. For practical reasons it is to be expected that all direct-connected steam-driven alternators of 500 Kv-a and larger, will be of this construction. However, for sizes smaller than 500 Kv-a, as proposed for destroyers and cruisers, it has been found to be more economical to use *salient-pole* slower speed generators, with speed reduction gears.

CHARACTERISTICS OF THE REVOLVING-WINDING SYNCHRONOUS GENERATOR.

In discussing the characteristics of revolving-winding syncronous generators, no distinction will be made between the revolving-armature and the revolving field types or the salient-pole and embedded-coil types. In general, the electrical characteristics are similar and any differences in fundamental construction are to be regarded as mechanical.

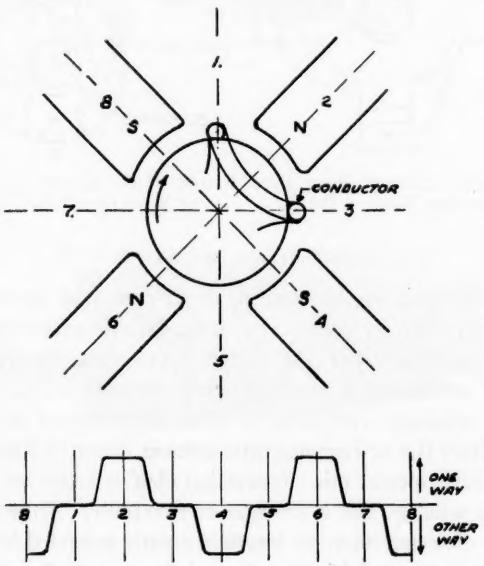


FIGURE 8.—GENERATION OF ALTERNATING E. M. F. IN A PLAIN DESIGN.

The Wave Shape.—When e.m.f. is generated in a synchronous revolving-winding alternator, it has been shown that the *frequency* of alternations is dependent on the R.P.M. of its shaft and the number of pairs of D.C. poles. The alternating form of e.m.f. or current, is due to the change in direction of the relation of the armature coils and the D.C. field flux when either one or the other is revolved by the power shaft. The wave shape of the generated e.m.f. and the consequent current, however, is not dependent on basic principles, but is dependent on the way in which the winding is distributed and by the way the flux spreads over the pole face. Ordinarily the wave shape of a plain design would appear as illustrated in Figure 8, which represents a smooth drum-armature core arranged to revolve within a set of poles which direct-current magnetizes the flux in the usual way. A wire secured to the outside of the drum would have alternating e.m.f. generated in it as shown in the plotted curve for eight positions of the wire.

By shaping the poles, altering the shapes of teeth, or modifying distances from armature core to pole faces at different places under the same poles, the distribution of flux across the pole face can be regulated to approximate a true *sinusoidal* wave e.m.f., as described in part one of this article. It is possible to conceive periodic e.m.f. waves as being triangles, sinusoids, parabolas, semicircles or rectangles. The mathematical ratio of effective to maximum values of these basic hypothetical curves are, respectively, .577, .707, .730, .835 and 1.000. However, in alternating current generation, the sinusoid or the sine-wave, is used as the basis of standard practice. This is for the reason that the true sine-wave is fundamentally free from *harmonics*. Physicists have proven mathematically, that all forms of curves, except the sinusoid, may be resolved into a sinusoidal curve with modifying harmonics. Then *per se,* all waves except sinusoidal contain more or less harmonics as part of their curve form. In the generation and application of alternating current, harmonics decidedly increase the losses in the metallic materials necessarily used as magnetic or conducting circuits in the design of power apparatus. Losses such as hysteresis and eddy currents, increase in proportion to the frequency of the current alternations. Where harmonics are pres-

ent, these losses are increased over those due to the basic plant frequency, in proportion to the frequency of the superimposed harmonics.

Voltage Characteristics.—When A.C. generators are required to supply constant-voltage circuits, which is the invariable requirement except for generators which supply ship's propulsion motors, special means are incorporated in the system to hold up the rated voltage as the current demands increase. Normally, as current is drawn from the generator, the volts supplied would drop like that of a shunt D.C. generator. If the voltage is to be maintained constant at the machine terminals, the exciting field current must be increased as the load increases. The drop in voltage under constant excitation is caused by the copper losses in the armature windings and the de-magnetizing effects of the armature current on the main fields. In the case of compound D.C. generators, the voltage is maintained by employing the main current as additional exciting current for the field.

In the case of the A.C. generator, the corrective means used for the D.C. generator are not practical as the main current is alternating. Earlier designs employed for this purpose a rectifying commutator in series with the main A.C. for compounding the fields. In addition, the power-factor of the main current, which in most installations is lagging and rarely exceeds 80 per cent, causes appreciable voltage loss, by the de-magnetizing effect of the lagging current which becomes more and more effective as the angle of lag increases. This effect is shown in Figure 9. Referring to the left hand sketch, the armature coil is in the position of maximum e.m.f. and if the current is in phase or at unity power-factor, the de-magnetizing effect is minimum. In the right hand sketch of the same figure, the coil is shown in the position it would be if the current were to be completely out of phase, or zero power-factor. The arrows show the relative direction of the current in the armature coil and the field winding. As they are opposing, the de-magnetizing effect is maximum because of the favorable position of the coil. The additional exciting current required to compensate for zero power-factor lagging current would be excessive. Conversely to the foregoing, if the current

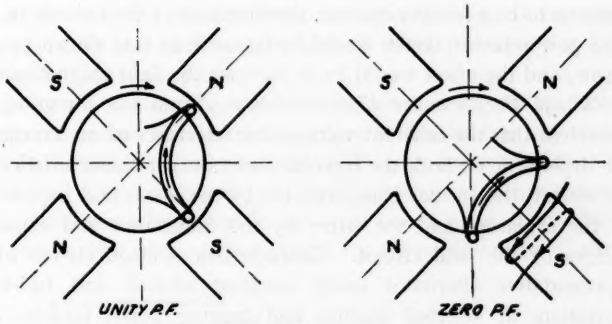


FIGURE 9.—MAXIMUM-CURRENT POSITIONS OF AN ARMATURE COIL AT UNITY AND AT ZERO LAGGING POWER-FACTOR.

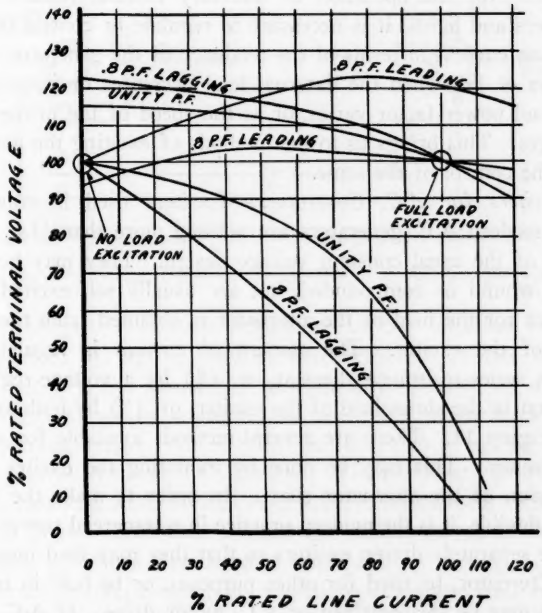


FIGURE 10.—CHARACTERISTIC CURVES OF CHANGE IN ALTERNATOR TERMINAL VOLTAGE UNDER CONSTANT NO-LOAD EXCITATION AND CONSTANT FULL-LOAD EXCITATION AT VARIOUS POWER-FACTORS AND LINE CURRENTS.

happens to be a *leading* current, the direction of the current in the zero power-factor sketch would be opposite to that shown by the arrow, and the effect would be to *increase* the field magnetization.

Voltage Performance Characteristics.—From the foregoing, it is evident that the inherent voltage characteristics of an alternator are dependent on both the internal and external characteristics of the circuit, the former being affected by the losses and impedence of the armature and the latter by the inductance and capacity factors of the load circuit. Characteristic voltage curves of a representative alternator under *constant* no-load and full-load excitation, at selected leading and lagging power-factors, are shown in Figure 10.

In order to obtain constant generator *terminal voltage* which is required for the operation of auxiliary motors, heaters, transformers and lights, it is necessary to regulate or control the D.C. exciting current in terms of the tendency of the generator voltage to rise or fall when the external load decreases or increases; as the load power-factor varies, or as the speed of the prime mover changes. This brings us to the methods of exciting the generator and the control of the same.

Exciters for A.C. Generators.—Exciters used in connection with modern A.C. generators are nothing more than D.C. generators of the usual constant voltage design. They may be either shunt-wound or compounded, but are usually self-excited. The current for the field of the alternator is obtained from the armature of the exciter. The amount of current is regulated by, (1) a series-resistance rheostat, or, (2) by a voltage-regulating rheostat in the shunt-field of the exciter, or, (3) by both systems. See Figure 11. There are several methods available for driving the exciter. This may be done by mounting the exciter on an extension of the alternator shaft. In order to make the system more flexible, it is the modern practice in commercial power-plants to use separately driven exciters so that they may feed more than one alternator, be used for other purposes, or be held in reserve. These may be steam-driven or A.C. motor drive. If A.C. drive, it is necessary to have a steam-driven set as a stand-by in case of complete failure of A.C. power. For Naval ships, it is pro-

posed to drive the exciters from the same turbines as the alternators, with extra load capacity in some cases, for supplying D.C. power for other purposes.

Automatic Voltage Regulators.—As shown in Figure 11, control of the alternator terminal voltage may be obtained to any degree within the range of the rheostats. This can be accomplished for loads that vary infrequently, by hand operation of either rheostat. For conditions which exist in the usual power plant, hand control is neither quick nor accurate enough for the requirements of main-

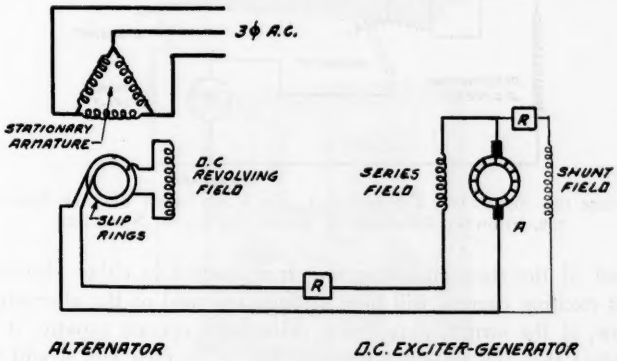


FIGURE 11.—SCHEMATIC DIAGRAM OF EXCITER AND ALTERNATOR CIRCUITS.

tained voltage. *Automatic* regulation of the terminal volts is accomplished by one or both of two methods. In one method (see Plate G), the rheostat is mechanically operated by a small reversing D.C. motor, connected to the exciter power by starting and reversing relays. These relays are operated by spring-loaded levers controlled by an A.C. solenoid. The movement of the solenoid is balanced when normal A.C. volts is maintained. The *time* element in this method is considerable and it serves to regulate only slow variations from normal volts.

In order to compensate rapid variations in voltage from normal, a second *high-speed* method is used independently or as a supplement to the motor-operated-rheostat method. There are several commercial types of high speed regulators, one of which is known

commercially as the Tyrell regulator. All these types operate by the same method which is explained by referring to Figure 12.

In the above diagrams, short-circuiting switches are shown, which if closed, cut out predetermined sections of the rheostat resistances. In the field circuit of the *alternator* this would allow higher exciting current to flow. In the exciter shunt-field circuit, it causes the exciter to generate higher voltage which in turn also circulates higher current in the alternator field. On the other

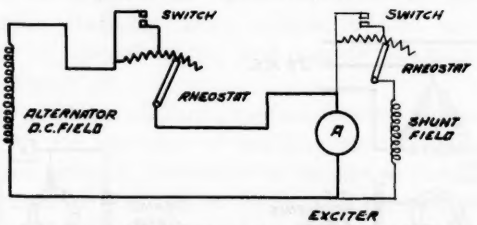


FIGURE 12.—SCHEMATIC CONNECTIONS OF A HIGH-SPEED VOLTAGE REGULATOR, SHOWING LOCATION OF SHORT-CIRCUITING SWITCHES.

hand, if the short-circuiting switch is opened in either rheostat, less exciting current will flow through the field of the alternator. Now, if the switch were to be closed and opened rapidly, it is evident that the resulting current, due to its time lag, would be of some value between the limits set by the closed and open values. The effective value of the current so controlled is regulated by the *time-element* of the switch operation. In general, *high-speed* regulators are designed to mechanically operate the short-circuiting contacts and the frequency of closing. The duration of the short-circuit is obtained electrically and mechanically by an electric clock mechanism or by an electric vibrator in conjunction with contacts operated by the varying torque or magnetic pull established by variations in the A.C. line voltage. The pulsating field current, which is to be expected with the high-speed control, does not cause appreciable pulusating A.C. generator voltage, because of the large electrical inertia factor of its field.

A Navy Type Automatic Voltage Regulator.—There has recently been submitted for Naval service an automatic voltage regulator which is designed to give the operating features of both

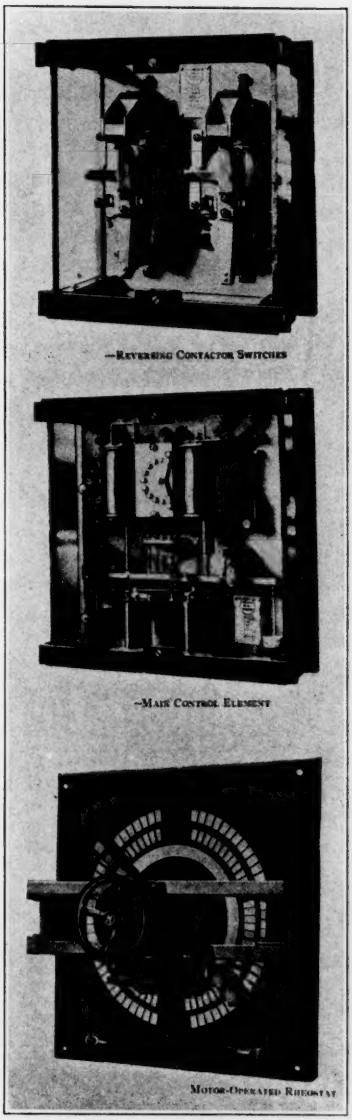
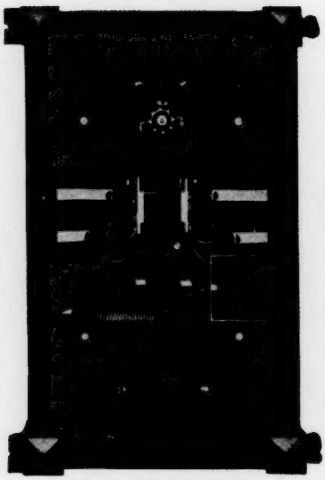
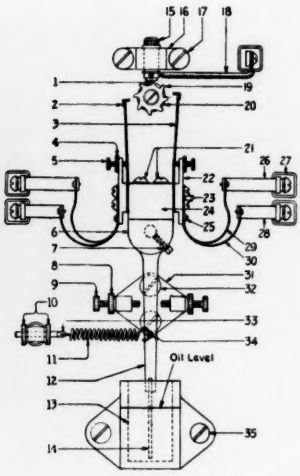


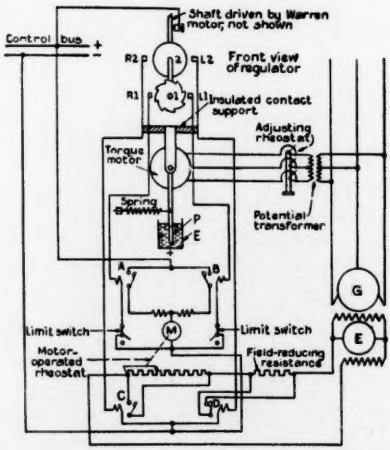
Plate G—Automatic Voltage Regulator Motor-Operated Rheostat Type.



Type FA-4 Voltage Regulator



Details of Type FA-4 Voltage Regulator



Elementary Wiring Diagram
Showing Series Field Rheostat Connection

PLATE H—AUTOMATIC VOLTAGE REGULATOR COMBINATION MOTOR-OPERATED RHEOSTAT AND SHORTED RESISTORS.

the low-speed and high-speed methods, combined in one device. Plate H shows the appearance of the control unit, and the accompanying wiring diagram shows the circuit arrangement of the operating parts. In this device, no vibrating mechanism is used. The equipment, however, is supplied with a continuously revolving double cam, attached to the shaft of a synchronous electric clock motor. Torque for deflecting the main lever arm is obtained from a three-phase induction torque motor, which is held stationary and stable at normal volts, by a counter-acting spring. Increasing or decreasing A.C. line volts allows the motor and the lever arm to deflect slightly one way or the other. One set of contact fingers mounted on the lever arm engage with rotating teeth of the clock driven cam. Another set of contacts mounted on the same lever arm engages a smooth portion of the clock driven cam. The double sets of contacts operate as follows: When the A.C. line volts change slightly, the torque motor deflects the lever arm and causes one set of contacts to be intermittently energized by contact with the clock cam. This in turn jogs the rheostat motor in the direction to bring the A.C. line volts back to normal. If the A.C. line volts are greatly disturbed, the lever arm deflects still further until the second set of contacts engage on the smooth part of the clock cam. This operation results in short-circuiting or open circuiting a portion of the field resistor which in turn results in a quick raising or lowering of the exciting current to establish normal A.C. volts. It is to be expected that in all forms of voltage regulators, if external load demand is applied in one step of relatively high percentage of the generator capacity, that voltage disturbance will be in evidence to the extent of at least 5 per cent for a period of at least 2 seconds, caused by the mechanical and electrical lag of the parts of the system. The voltage regulation under nominal load changes is expected to be within plus or minus one per cent.

Unbalanced Phases, Voltage Regulation.—In the foregoing description of the means of obtaining generator normal terminal voltage, it is assumed that if the generators are three-phase, the external load is balanced between the three phases as would be expected in a three-phase motor load or where single-phase loads are distributed and operated equally between each of the phases.

Equal voltages then would be available between terminals 1 to 2, 2 to 3 and 3 to 1 of a three-wire system. However, in most installations, it is impracticable to either balance the distribution of single-phase power or to maintain the balance under all operating conditions. Unfortunately, there is no means available of increasing the field excitation of one of the phases to compensate for an increased load demand, without also increasing the excitation of the other two phases. This is because the excitation is common to all three phases. Therefore, in unbalanced distribution, the more heavily loaded phase or phases will be at a lower voltage than the remainder. Whether the unequal voltages are higher or lower than normal would depend on which phase is used to indicate the operating conditions or which phase is used to control the automatic voltage regulator. Automatic regulators operating on the simultaneous voltage effect of the three-phase would serve to produce an average of the voltage of the three unbalanced phases. In this case, the terminal voltage of the more heavily loaded phases would be below the average voltage, and the lightly loaded phases would be above the average. The amount of variation from the average would depend on the extent of the unbalance and the characteristics of the machine. The voltage unbalance between phases of an unbalanced system may be of the order of 3 to 5 per cent. In commercial distribution, it is customary to install external automatic phase voltage balancers in each phase. These devices usually consist of relatively heavy series-wound inductors in which the magnetic angular relation of two coils is shifted by a motor driven control. Higher or lower voltage is induced in the separate phases, to maintain the average nominal voltage.

PARALLEL OPERATION OF SYNCHRONOUS GENERATORS.

Alternating current generators of the synchronous type are connected in parallel to serve constant voltage circuits, for the same reason that D.C. machines are so used. That is, for the purpose of increasing the plant capacity beyond the range of a single unit; to serve as additional reserve power for expected demands; or, to permit shutting down of one machine and cutting-in of a stand-by machine, without interrupting the voltage or current being supplied to the bus-bars. Although A.C. generators

are readily paralleled with the aid of modern auxiliary devices, nevertheless, it is more difficult to put them properly in parallel, for the reason that they generate *alternating* e.m.f. The e.m.f. on a 60-cycle generator reverses every 1/120 second, and if the circuits are closed when the e.m.f. is the wrong direction, results would be practically a short-circuit at a voltage of the bus-bars plus that of the incoming generator; the mechanical shock to the windings and machine structure would be severe if not disastrous.

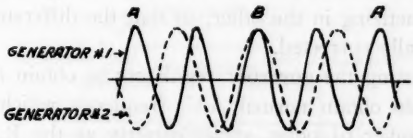


FIGURE 13.—PERIODIC TIME RELATION OF THE PHASE VOLTAGES OF TWO SEPARATED GENERATORS RUNNING AT SLIGHTLY DIFFERENT SPEEDS.

Comparison of the voltage curves of one of the phases of two three-phase generators operating independently but at slightly different speeds is indicated in Figure 13. At the *time* periods indicated by the points "A," the short-circuit condition would result if the machines were to be connected in parallel. At the point "B," they may be paralled. The process of paralleling two machines, therefore, may be listed step by step as follows:

(1) Adjusting the D.C. fields to obtain practically normal A.C. voltages.

(2) Adjusting the governor regulators of the prime movers to obtain normal speed.

(3) Observing the time period relation of the maximum voltage points of one phase of each generator, by means of a *synchronizing* device.

(4) Raising or lowering the speed of one of the generators until the time periods slowly coincide.

(5) Closing the paralleling switches at one of the periods of coincidence.

(6) Adjusting the speed governor of the prime movers until the load is suitably shared and normal frequency is established.

(7) Adjusting the D.C. field rheostat of both machines, to obtain normal combined parallel voltage.

The reasons for the various paralleling steps and the limitations of what is termed *normal* adjustments are as follows:

(1) Voltage equalization of the two machines before paralleling does not have to be as precise as for D.C. generators. Variation between machines to the extent of 3 to 5 per cent would be considered reasonable. The effect of higher than normal voltage on an incoming machine would be to circulate currents around the local circuit and the A.C. armatures, due to the difference in e.m.f. These reactive currents are magnetizing in one armature and de-magnetizing in the other, so that the difference in voltage is automatically corrected.

(2) Adjusting the governor regulators to obtain *normal speed* is required to obtain normal A.C. frequency which, on a fixed designed number of poles, varies directly as the R.P.M. of the rotor. The speed control is usually accomplished by the use of a small D.C. motor, energized from the exciter current and operated at the switchboard. The motor serves to mechanically lengthen or shorten the governor spring.

(3) To effectively observe the *time* period relation of the A.C. voltage of the two machines before paralleling requires the use of one of the several synchronizing devices. The simplest form is in the use of two incandescent lamps of voltage rating comparable with the generator voltage. If the generator voltage is too high for commercial lamps, additional lamps in series or transformers are used to prevent overvoltage on the indicating lamps. Synchronizing of three-phase generators is always done on one of

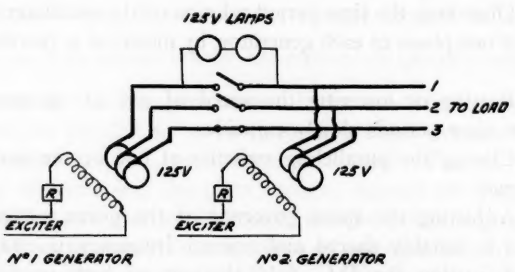


FIGURE 14.—SYNCHRONIZING, USING 2 LAMPS AS INDICATORS, APPLIED TO A REVOLVING-ARMATURE TYPE GENERATOR.

the phases only, assuming that on the *initial* tryout the proper relation of the wiring hook-up of the other phases has been established. Figure 14 shows a diagrammatic paralleling hook-up, using synchronizing lamps on one phase; Figure 15 shows the use of transformers for higher voltage generators.

(4) It is evident in the lamp circuits shown in Figures 14 and 15, that when the e.m.f. of the generators is at the maximum and of opposite polarity, both lamps will light at full voltage. Assum-

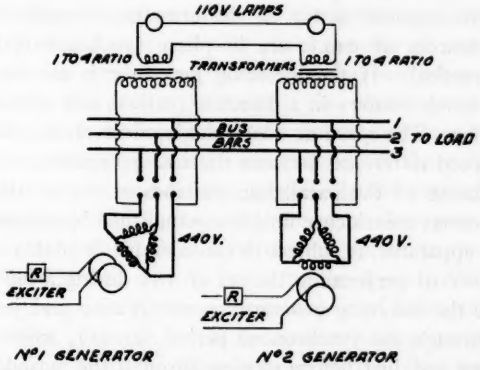


FIGURE 15.—SYNCHRONIZING, USING TRANSFORMERS AND 2 LAMPS AS INDICATORS, APPLIED TO REVOLVING-FIELD TYPE GENERATORS.

ing that the generators are running at slightly different speeds, if the slower one is speeded up, or if the faster one is slowed down, there will come a period of time, lasting a few seconds, when they will pass through a coincidence phase where the e.m.f. will be maximum and the same polarity. This will gradually change to the opposite relation. During the period of the same polarities, the lamps will have no voltage and will be dark. The generators may be connected in parallel at this time. It is to be noted when using transformers as in Figure 15, by cross-connection of the transformer leads, either intentionally or otherwise, the lamps may be caused to *light* at the synchronous period. On first installation, it is customary to establish whether the dark or light period is the indication of synchronism. The use of lamps for synchronizing has, in recent years, been replaced commercially by a device

known as a *synchroscope*. Although the lamp method is valuable as a stand-by, for the reason that it can be readily assembled upon failure of the synchroscope, the objection to the lamp method is that it does not indicate which generator is the faster or the slower. This must be determined by trial and error. The *synchroscope* is a clock-like device with a single indicating hand. The hand is mounted on the shaft of a split-phase motor armature, excited from the voltage of one generator. The other generator or the bus-bars, excites a two-pole field structure enclosing the armature. The electro-magnetic action of the armature is such that when the two sources of e.m.f. are in phase synchronism, the hand remains vertical. If the incoming generator is too fast or too slow, the hand revolves in a direction marked fast or slow as the case may be. The *speed* at which the hand revolves indicates the relative speed difference between the two generators.

(5) Closing of the paralleling switches at one of the periods of synchronous coincidence unless accomplished by automatic synchronizer apparatus, is subject to considerable flexibility in regard to the choice of performing the act at, (*a*) during a *lull* moment, (*b*) while the incoming generator is *accelerating* and just before passing through the synchronous period; or (*c*), while it is *de-accelerating* and just before passing through the period. Choice (*a*) requires very fine control of the generator speed and is uneconomical in regard to the time expended waiting for the period of stabilization. Choice (*b*) is the usual one, as the inertia of moving parts and the steam flow is more favorable to receive the load without shock if the machine is accelerating.

(6) Adjusting the speed of the prime movers by means of the hand throttle or, as is usual, by external adjustment of the governor control, leads to two results. Assuming that the two machines are in parallel, are of the same design, are of equal rating and are sharing the external load, changing the speed control equally on both machines will raise or lower the A.C. cycles at the bus-bars and will also change the voltage proportionally. If one of the machines loses power or torque by operating the speed control, by failure of the governor or by lack of motive power, the remaining machine, if within its power capacity, will continue at the same frequency and voltage (within its regulating capacity).

At the same time, the load output of the one machine will reduce in proportion to its power failure, and the other machine will increase its output until the bus-bar is satisfied. If the power failure is increased on the one machine to a sufficient extent, it will continue to run at synchronous speed as a *synchronous motor,* absorbing sufficient bus-bar current to overcome its electrical and mechanical losses.

(7) Adjusting the D.C. field rheostats of both A.C. generators to obtain normal bus-bar voltage is greatly different in effect, from two D.C. generators in parallel. In the latter case, variations in the rheostat adjustment will also cause the current output to the bus-bars of one machine to increase or decrease while the other machine will deliver the difference. In other words the field rheostats of *direct-current* generators are not only used to regulate the delivered voltage, but are used to divide or share the bus-bar current demand, to any desired degree. In the case of the *A.C.* generators, variation in the field rheostats does not effect the sharing or dividing of the *useful* current output to the bus-bars. This is accomplished only by operating the speed controls. Raising or lowering of the rheostat adjustment of both generators at the same time, raises or lowers the bus-bar voltage proportionally. If the rheostat of *one* machine is lowered, the other machine will circulate an out-of-phase current, *e.g.* a wattless current, which will be indicated as a change of current on the individual output ammeters of the generators, but if wattmeters are installed in each circuit they will indicate practically no change. The out-of-balance rheostatic adjustment, however, due to the wattless circulating current, will be evidenced in the out-of-balance indications of power-factor meters if installed as part of the switchboard equipment. Equalization of the current flow to the *D.C. fields* of the two machines, as evidenced by ammeters installed in these circuits, also furnishes a guide as to probable balanced generated voltages.

THE EFFECT OF POWER-FACTOR.

One of the important differences between the generation and application of alternating-current and direct-current is the item of *power-factor* or the phase angle of lag of the current behind the e.m.f. With direct-current this item is unity, *e.g.* 100 per cent,

and therefore no problems of generation or transmission are encountered from the power-factor aspect. With alternating-current the lagging current not only de-magnetizes the generator field with resultant increase in the field exciting current required to maintain the terminal voltage, but, the increased current required to produce effective power at lowered power-factor causes additional copper losses in the generator, cables, bus-bars, switches and windings of the power apparatus.

The Effect of Power-Factor on the Exciter Requirements.—In order to obtain an exciter of suitable capacity for the generator requirements, it is necessary to make a survey and estimate probable power-factor to be expected from the power demand for which the installation is required. Normally, the power-factor of most installations is tentatively placed at 80 per cent. In the absence of information or survey, commercial generators are usually rated on the basis of 80 per cent power-factor, lagging. Figure 16 illustrates the rapid change in the power requirements of

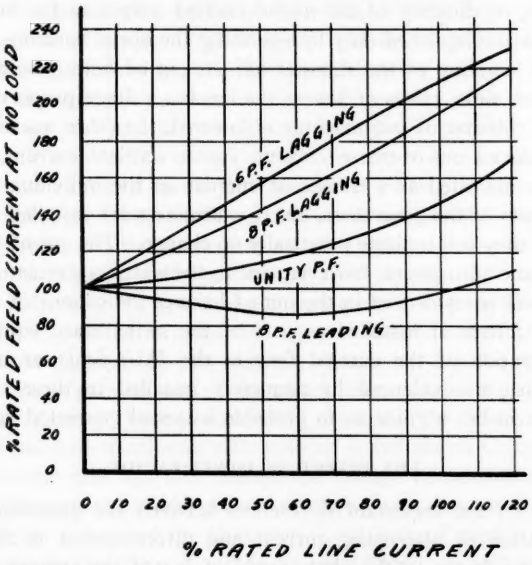


FIGURE 16.—CHARACTERISTIC CURVES OF FIELD EXCITATION AT VARIOUS LINE CURRENTS AND POWER-FACTORS TO MAINTAIN CONSTANT NORMAL LINE VOLTAGE.

an exciter, for small per cent changes in power-factor. If the plant power-factor is lower to any considerable extent than the normal rating, it may result in overloaded exciters and overheated generator field windings.

The Effect of Power-Factor on Plant Losses.—The effect of the increased current which is obtained with lowering plant power-factor is quite marked on the basis of I^2R copper losses in all parts of the circuit. These losses, which are additional to the normal copper losses of a direct-current installation, are evidenced in additional heat, in proportion to the *square* of the excess current ($W = I^2R$), and in voltage loss in *direct* proportion to the excess current ($E = IR$). In installation where it is expected that the full-load power-factor can not be maintained at a reasonable value, it is usual to provide oversize copper circuits, switches, circuit-breakers, etc., so that the heating and voltage losses may be reduced to a minimum by lowering the resistance (R) factor of the equations. The marked increase in losses with a reduction in power-factor above the losses occurring at unity

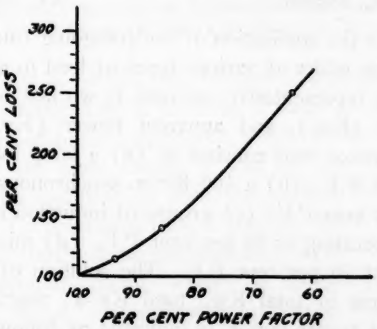


FIGURE 17.—CHARACTERISTIC GENERATOR AND DISTRIBUTION PER CENT LOSSES AT VARIOUS PLANT POWER-FACTORS.

power-factor is shown in Figure 17. The curve is based upon constant Kw. and applies to all current-carrying parts. Inspection of the curve shows that at 80 per cent power-factor, the loss is 156 per cent of the unity power-factor; at 70 per cent power-factor, the loss is 205 per cent of the unity power-factor loss.

Effect of Power-Factor on Current Values.—In any alternating-current circuit excepting where the power-factor is unity, the

current flow may be said to be composed of two currents, one being in phase with the e.m.f., known as the *energy* current, and the other being 90 degrees out of phase with the energy current, known as the *quadrature* or *reactive* current. The *resultant* current is the vector sum of the energy and quadrature current, mathematically expressed in the solution of the long side of a right-angle triangle as, I resultant $= \sqrt{I^2 \text{ energy} + I^2 \text{ quadrature.}}$ The above relations are shown graphically in Figure 18, or as in Figure 19 in terms of both volts and current (Kw. and Kv.-a.). The *power-factor* of the system is the ratio of *energy* current (or Kw.) to the *resultant* current (or Kv.-a.).

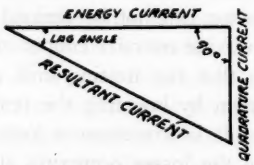


FIGURE 18.—CURRENT RELATION IN AN A.C. SYSTEM.

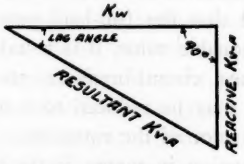


FIGURE 19.—POWER RELATION IN AN A.C. SYSTEM.

To illustrate the application of the foregoing fundamentals, and to illustrate the effect of various types of load in an A.C. system, the following representative example is worked out in terms of actual power (Kw.), and apparent power (Kv.-a.). Assume that the alternator load consists of (*a*) a lamp load of 200 Kw. at 98 per cent P.F., (b) a 100 Kv.-a. synchronous motor operating at 100 per cent P.F., (*c*) groups of induction motors totalling 400 Kv.-a. operating at 80 per cent P.F., (*d*) miscellaneous load of 154 Kw. at 70 per cent P.F. The solution of the above example, in terms of total Kw., total Kv.-a., reactive Kv.-a. and resultant plant power-factor, is indicated as follows:

	Kv.-a.	*P.F.*	*Kw.*	*Reactive Kv.-a.*
(*a*) Lamps	204	.98	200	$\sqrt{204^2 - 200^2} = 41$
(*b*) Synchronous motor	100	1.00	100	$\sqrt{100^2 - 100^2} = 0$
(*c*) Induction motor	400	.80	320	$\sqrt{400^2 - 320^2} = 240$
(*d*) Miscellaneous	220	.70	154	$\sqrt{220^2 - 154^2} = 157$
Totals	924		774	438

From the above, the resultant power-factor is $\frac{774}{924} = .837$ or 83.7 per cent. In Figure 20, the angular relations, the reactive values and the resultant factors are plotted to indicate the individual and combined load items of the example.

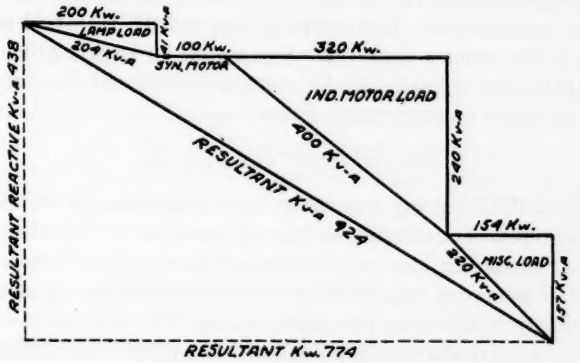


FIGURE 20.—GRAPHIC ANALYSIS OF DIFFERENT COMBINED TYPES OF A.C. PLANT LOADS IN TERMS OF KW. AND KV.-A.

Corrective Kv.-a.—From the foregoing example, the reactive Kv.-a., which totals 438, is used as a basis in determining the amount of out-of-phase component which would be required to be applied to the plant circuit if 100 per cent power-factor is desired. In the load problem cited, the reactive Kv.-a. is a *lagging* component. To neutralize this, a *leading* 438-Kv.-a. component in the form of electro-static condensers, or in the form of an over-excited synchronous motor, would be necessary to supply the compensating Kv.-a. As indicated in item (*b*) of the example, the use of *synchronous* motors as part of the plant load does not necessarily improve the power-factor, as they normally are rated in current capacity and field excitation limitations to deliver only its own 100 per cent power-factor. In other words, its Kw. rating equals its Kv.-a. rating. Of course, the addition of any high power-factor load improves a low plant power-factor by virtue of the reduction in the *preponderance* of the low power-factor load. To use a synchronous motor for plant power-factor correction

in addition to using the motor for driving a mechanical load would require a type of motor known as a *leading* power-factor, synchronous motor. In this case, the problem would consist of determining and adding the horsepower item in terms of Kw. to the plant load, and then again calculating the total resultant reactive Kv.-a. to produce 100 per cent (or some other selected per cent) plant power-factor. The current rating and the D.C. field excitation of the motor is then specified as a basis of designing the motor so that it may operate both the mechanical load and the corrective Kv.-a. within its temperature limitations.

SUMMARIZING CONCLUSIONS.

From the foregoing description and explanations of the various basic types of alternating-current generators, it is evident that there are one or more predominating characteristics inherent in each of the types, and which are not in common, except as to the possibility of delivering sine-wave current. The choice in the selection of a particular type for shipboard use, therefore, is influenced by its outstanding characteristics and the machine sizes available in practical commercial designs. On the above basis, the adaptability of the various types is outlined as follows:

(1) *Induction Generators.*—Available in a large range of sizes from 10 Kv.-a. to 10,000 Kv.-a. and upward. Not particularly applicable to shipboard use as they must be used in conjunction with predominating synchronous alternators.

(2) *Inductor Generators.*—Available in small sizes only, up to 10 Kv.-a. Formally applicable for generating 500-cycle current for radio power. Now replaced by vacuum tubes operated on 60-cycle power.

(3) *Revolving-Armature Generators.*—Available in small sizes only, up to 25 Kv.-a. Applicable for auxiliary communication or control service. Usually driven by a D.C. motor in order to employ the D.C. current for compounding purposes.

(4) *Inverted Rotary Converters.*—Available in fractional-power sizes and in large sub-station sizes. Applicable in small sizes to D.C. ships, as a convenient source of A.C. current to operate salinity indicators and similar independent A.C. operated devices.

(5) *Revolving-Field Generators, Salient-Pole Type.*—Available in small to large sizes, ranging from fractional-power to 500 Kv.-a. Applicable for shipboard use as the main source of A.C. supply where the prime-mover is, of necessity, the reciprocating type geared turbine or other forms of slow speed drive. The salient-pole construction lends itself to the use of a large number of poles so that 60-cycle current may be generated at speeds as low as 90 R.P.M. for an 80-pole design.

(6) *Revolving-Field Generators, Embedded-Coil Type.*—Available in medium to very large sizes, from 500 Kv.-a. to 10,000 Kv.-a. and larger. Applicable for shipboard use as the main source of A.C. supply, where the prime-mover may economically operate at 3600 R.P.M. as a 2-pole, 60-cycle generator, or at 1800 R.P.M. as a 4-pole, 60-cycle generator. The number of poles in revolving-field, embedded-coil type generators is limited practically to either 2 or 4. For 60-cycle generation, this limits the normal prime-mover to a choice of 1800 or 3600 R.P.M. unless reduction gears are to be considered. Prime-mover speeds of 1800 or 3600 R.P.M. precludes the use of a reciprocating form of drive, and in general the steam turbine is invariably used on shipboard for this type of generator. For *main propulsion* A.C. power, where variable frequency generators of this same type are used, turbines are operated at varying governor controlled speeds, in which case the frequency is proportional to the operated speed in the same ratio as at the 60-cycle, 2-pole, 3600-R.P.M. or the 60-cycle, 4-pole, 1800-R.P.M. generators. For example, a 2-pole generator operated at 1800 R.P.M. would deliver current at 30 cycles or a 4-pole generator operated at 900 R.P.M. would also deliver 30 cycles. Therefore, in general, *variable* frequency main-propulsion generators follow the same fundamentals and may be of the same basic types as those used for *constant* frequency power distribution.

HIGH PRESSURE STEAM JOINTS AND GASKETS.

By Lieutenant Ralph C. Kephart, U. S. N., Member.

It has been in but comparatively recent years that the economical advantages of high steam pressures and temperatures have been recognized and put into practice. Shore plant practice in this respect has gone well ahead of marine practice. However, there are many sound reasons for this, as a marine installation presents many problems not found in shore plants. Fundamentally, these problems arise from the lack of space and rigid foundations, the requirements for ease of maintenance, and the necessity for cutting down weight as much as possible.

The results of investigation of two of these problems will be presented here. First, new methods of making up joints on small steam lines (up to and including ¾ inch), such as gage and drain lines, to replace screwed connections and flange connections. Second, new types of gaskets to supersede the ordinary asbestos sheet gasket.

The subject of small joints will be taken up first. Figure 1 shows the types of joints evolved in the Bureau of Engineering and submitted to the Engineering Experiment Station at Annapolis for test. The nipple union and double union joints used conical seats with the flared end of the tubing acting as a seat. The single union has a conical seat with the tubing welded to the fitting inside and outside. The flange joint has the tubing rolled in, the inside of the flange being grooved. Gaskets are used only with the flange, which was included in the tests as a standard of comparison.

Figure 2 shows a modified union, using a gasket, with the tubing electric arc welded to the fitting. Figure 3 shows a standard American Railway Association fitting modified to the extent that the fitting is bored out to receive the tubing which is also arc welded to the fitting. This fitting uses no gaskets, one seat being conical, and the other rounded to the arc of a circle.

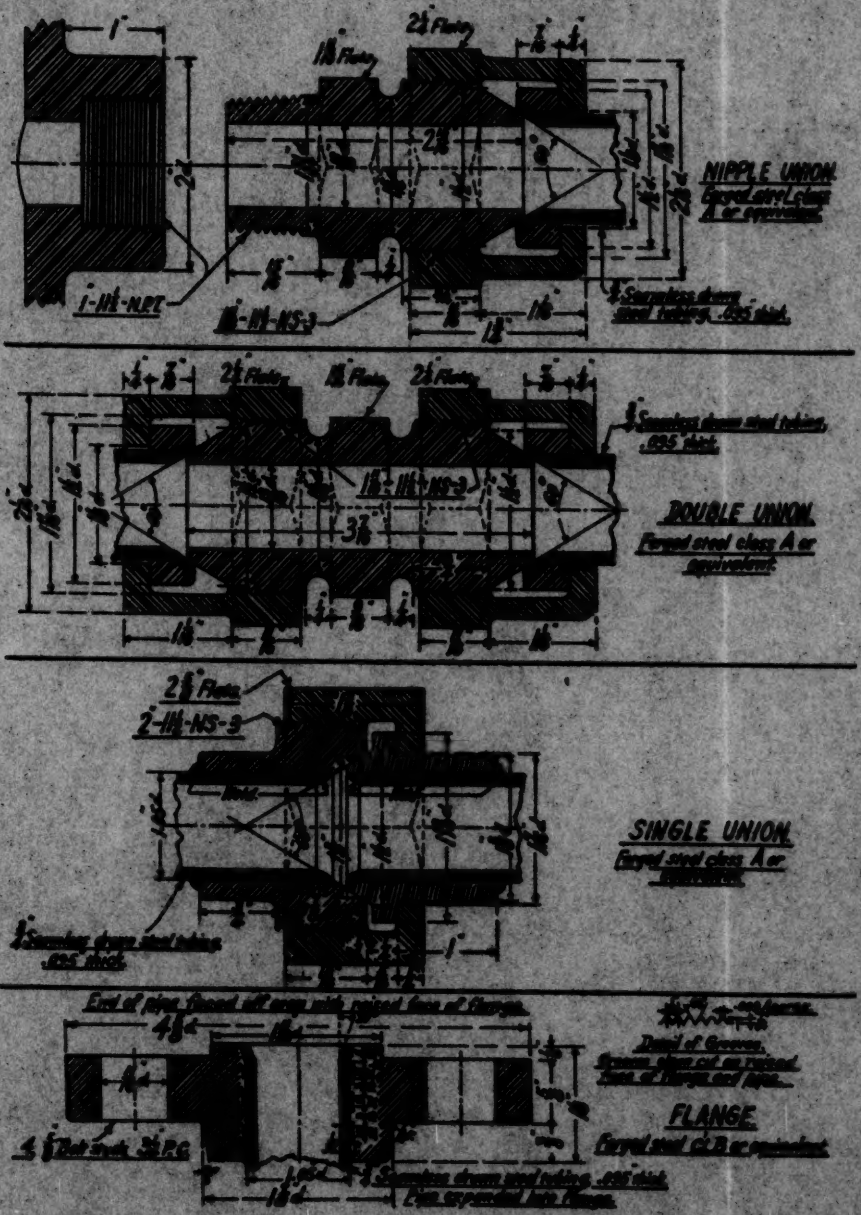

NIPPLE UNION
Forged steel class A or equivalent
DOUBLE UNION
Forged steel class A or equivalent
SINGLE UNION
Forged steel class A or equivalent
FLANGE
Forged steel C.I.B or equivalent

FIGURE 1.

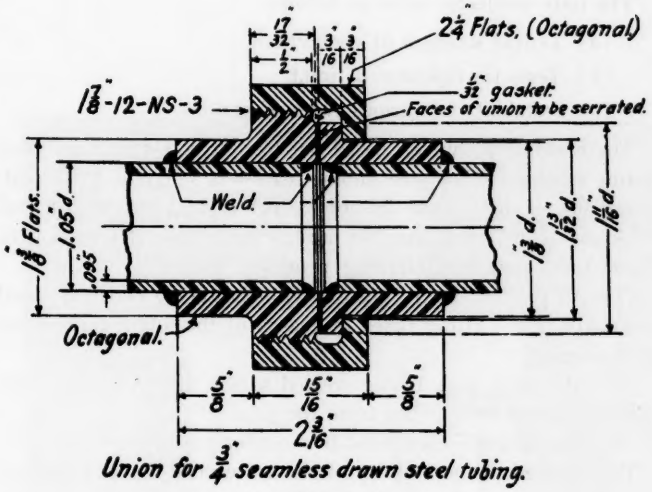


FIGURE 2.

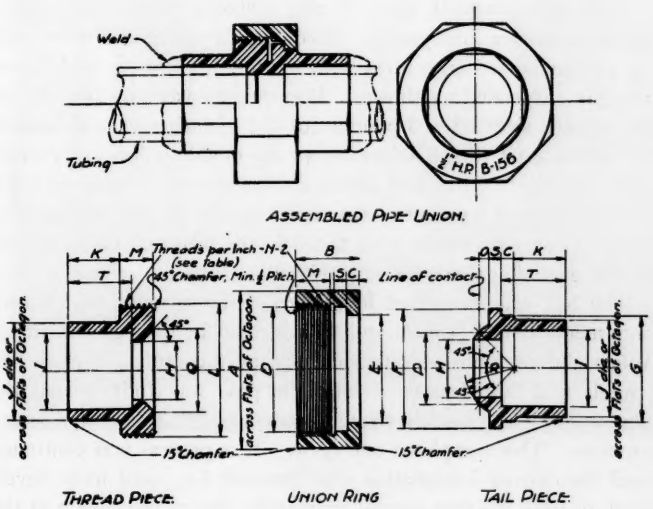


FIGURE 3.

The tests conducted were as follows:

(*a*) Tensile strength of connections.

(*b*) Tests for tightness of joint.

(*c*) Suitability of various types of gaskets.

The tensile tests were carried out to destruction in a standard testing machine. The steel tubing used was standard I.P.S. outside diameter in nominal sizes of ½ inch and ¾ inch with a wall thickness of .095 inch, and the fittings were Class B forged steel. These tests gave the following results:

(1) All the unions in Figure 1 with the tubing ends expanded and flared in the fitting failed by reason of the tubing pulling out of the fitting.

(2) All joints with tubing welded to the fitting failed in the tubing 2 inches to 7 inches from the joint.

(3) The flanged joints failed in the tubing as in (2).

This showed the superiority of the arc welded construction for the union type of joint, as the welding was stronger than the tube itself.

The next group of tests (*b* and *c* above) were made with a layout as shown in Figure 4. Two parallel assemblies were made up and supported on centers with the test joint in the middle and weights suspended as shown. The weights were 61 pounds for the ½-inch size and 103 pounds for the ¾-inch size, with centers 66 inches apart for all joints except the double union. The latter was tested with centers 46 inches apart and with weights of 107.8 pounds and 182.2 pounds respectively for the ½-inch and ¾-inch sizes. Gasket materials were tested in the inlet and outlet flanges of the assemblies concurrently with the joints under test.

The test was conducted by putting water through one branch and steam at 600 pounds and 700 degrees F. through the other. When the pressure and temperature at the outlet reached 600 pounds and 700 degrees F. and the pipe was fully heated the flow in each was quickly shifted from water to steam, and steam to water. This completed one cycle. This process was continued until the desired information was obtained, *i.e.*, until leaks developed, or until the pipe sagged excessively due to weakening at the

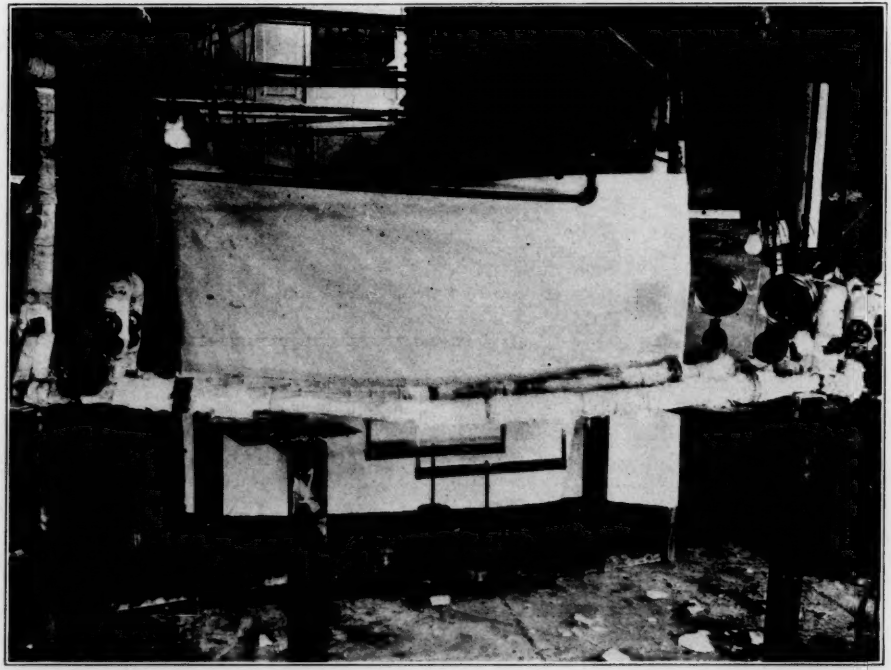


FIGURE 4.

elevated temperature. Steam for the test was supplied by two 600-pound pressure experimental boilers, and passed through a superheater coil in an oil fired furnace.

The results of the tests for tightness of the joints showed that the A.R.A. type, Figure 3, and single union, and flange types of Figure 1 were most satisfactory. These three joints withstood more than 100 cycles without leaking either steam or water. The double union, Figure 1, showed appreciable steam leaks, while the modified single union using a gasket, Figure 2, leaked on some runs and not on others, depending on the gasket material. The final choice of joint lay between the first two just mentioned. The factors affecting the choice were weight, ease of installation and ease of maintenance. The A.R.A. type (Figure 3) was the lightest, weighing but 1.35 pounds; the single union type (Figure 1) was next at 2.55 pounds. The flange was heaviest, weighing 7 pounds, including four bolts. Insofar as making up the joints is concerned, there is very little choice if electric welding equipment is available, and it is on all ships except destroyers, in which case tenders are available. Maintenance troubles should be practically nil.

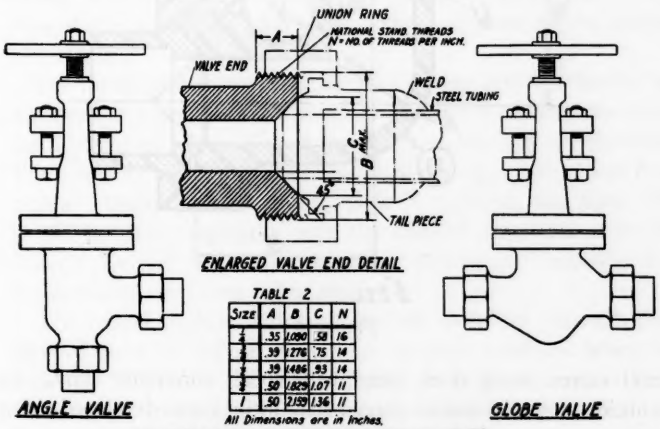


TABLE 2

Size	A	B	C	N
¼	.35	1.090	.58	16
⅜	.39	1.276	.75	14
½	.39	1.486	.93	14
¾	.50	1.829	1.07	11
1	.50	2.159	1.36	11

All Dimensions are in Inches.

THE TYPE BONNET AND STUFFING BOX GLAND SHOWN ARE FOR ¾" AND 1" VALVES. ¼", ⅜" & ½" VALVES HAVE UNION BONNETS & STUFFING BOX GLANDS ADJUSTED BY A PACKING NUT.

FIGURE 5.

The A.R.A. joint having proved to be the most satisfactory in the union connection, the next step was the development of standards adapting this type of joint to valves and fittings. The results are shown in Figures 5 and 6. As mentioned before, the union was a standard commercial 600-pound fitting, bored out to receive the tubing for welding, in lieu of the threaded connections. By threading and seating the valve ends similar to the thread piece in Figure 3, it was possible to use the standard union ring and tail-piece of Figure 3 for the valve connections. Figure 5 shows this adaptation on a standard line of 600 pounds pressure forged

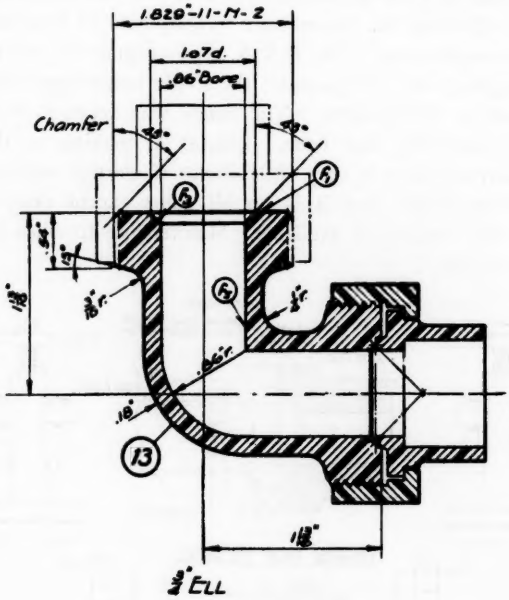


FIGURE 6.

steel valves, using discs faced with cobalt chromium alloy. In addition, a special line of forged steel fittings was developed using this type of joint. Figure 6 shows a typical ell on Bureau standard plan 3-S-376. In all cases, the complete valve or fitting includes the necessary number of union rings and tail-pieces ready

for use. The standards as finally developed went up to and included the 1-inch size of tubing.

In the investigation of gaskets, the following materials were tested:

1. Soft steel.
2. Soft aluminum.
3. Soft copper.
4. Soft monel.
5. Copper with asbestos insertion.
6. Asbestos.

The metals used for gasket material were annealed so as to obtain the desired softness. As mentioned before, the gasket materials were tested in the flanged inlet and outlet joints at the same time the experimental joints were being tested. As a result, it was found that the monel gasket was superior in all respects, with copper as second choice.

The asbestos and the copper-asbestos gaskets were definitely eliminated for service where considerable water would be encountered in the lines, which condition would obtain in these sizes of piping, considering their probable use for drains, gages, etc. They showed frequent leaks, and were unduly softened by the action of the water.

The metal gaskets were satisfactory during this test insofar as leakage was concerned, but were not all equally satisfactory from the viewpoints of erosion, corrosion, removal and replacement. The aluminum gaskets showed signs of pitting, erosion and corrosion. Under service conditions it is believed that their life would be short, especially with the alkaline condition of boiler water. Neither were the aluminum gaskets in condition for replacement, new ones being required.

The copper gaskets withstood the test well, and although they showed signs of tarnish, they were in good condition when removed. Practically the only objection is their extreme softness on removal, so that they distort easily, making it difficult if not impossible to replace them. However, copper has the advantage of availability on all ships, and is easily worked with simple tools.

The soft steel gaskets retained their strength but were corroded, due principally to the intermittent periods of water flow, and so considered undesirable.

The monel gaskets retained their strength and could be removed and replaced easily. They were always found to be in excellent condition upon removal, although they showed signs of tarnish in some cases. As stated before, the monel proved to be the best gasket material of those tested, with copper as an alternate satisfactory material where monel was unavailable. As a result of the gasket tests, monel will be used for gaskets in all new construction.

FURTHER DISCUSSIONS ON NOTES ON E.H.P. CALCULATIONS.

REPLY TO DISCUSSION BY LIEUT. H. A. SCHADE (CC), U. S. N.

By HENRY F. SCHMIDT.

In regard to the difficulty of making model tests, I am of course aware that the frictional coefficients can be quite accurately corrected for the scale effect, but the corrections for scale effect as influencing the resistance, do not, at least directly, apply to the corrections which it would be necessary to make for the scoops.

The thickness of the frictional belt is much greater on the model proportionately than on the ship, with the result that the quantity of water circulated, as well as the kinetic energy imparted to it, would be seriously influenced in the model, and the knowledge existing of the variation of velocity with distance from the skin in the frictional wake is hardly sufficient to make accurate corrections from model to full scale.

I was aware that the Model Basin had at one time attempted to measure the resistance of scoops. As I remember, a scoop pipe and discharge, making an angle of about 25 degrees with the skin, was installed on the model, and the water was permitted to flow freely through it, so that the velocity with which the water was discharged rearwardly from the overboard discharge was substantially equal to the velocity of the frictional wake belt, that is, the velocity at which it had been taken in. Under these circumstances, as Lieut. Schade points out, the direct effect would only be the additional frictional effect of the pipe plus the wave making, and additional friction due to stream-line disturbances caused by the removal and replacement of the water from the frictional wake belt. This does not represent the loss caused by an actual scoop.

In the last paragraph, page 534, of the November JOURNAL, Lieut. Schade says: "In the first place, the circulating water is

taken not from water at rest," etc. I anticipated this thought in the discussion of Commander Saunders' and Lieut.-Commander Pitre's paper,* " Full-Scale Trials on a Destroyer." I said the following:

" It may be thought that since the water which enters the scoop or sea chest has had imparted to it a certain fraction of the ship's velocity before it arrives at the entrance, it is not correct to take the ship's velocity in calculating the H.P. loss. This is not true, however, since when the layer of water adjacent to the ship is removed, this is replaced by an equal volume of water from farther out, which must then acquire the same fraction of the ship's velocity when it comes into contact with the hull plating, thus compensating for the difference in velocity.

" Further, the disturbance of the stream lines caused by this change of flow condition is the reason for the factor ' F.' "

In the same paragraph, Lieut. Schade says: " Secondly, the circulating water cannot acquire the full velocity of the ship; otherwise, there would be no motion of the water relative to the ship and no circulating water would flow." I think Lieut. Schade has neglected the fact that the function of the scoop is to create a difference of pressure in order to force the water through the condenser and that the scoop is connected to the condenser water box, the area of which is so large that the water is brought substantially to rest in a fore and aft direction. It is the slowing up of the water relative to the scoop and piping system which converts the velocity of the water relative to the ship into the necessary static pressure required to overcome the resistance through the condenser tubes, that is, when the water is in the larger section of the water box, it is substantially at rest with respect to the ship and has therefore imparted to it the full forward velocity of the ship. The velocity through the overboard discharge pipe is generally of the order of 8 to 12 feet a second, and as this is discharged at an angle of about 30 degrees, the longitudinal component of this velocity is about 9 feet a second.

In the case of the example cited, 42 knots corresponds to approximately 71 feet per second through the water, so that the ratio of the energy in the water discharged relative to the ship compared to the energy required to pick it up and accelerate it is 9^2 to 71^2,

* Extracts of which may be found on page 131 of the Notes in this issue.

or approximately 1½ per cent of the total energy imparted to the water by the ship, and for the case of an inclined overboard discharge, my formula is in error to this extent. However, the disturbance caused by the displacement of the stream lines by the overboard discharge is many times the recovery due to this relatively slight sternward velocity.

This also answers the third argument. I am in perfect agreement with Lieut. Schade that the effect of scoops does augment the resistance of the ship both by its effect on the frictional belt and by its effect on the wave-making component. The latter, however, is probably a very small percentage of the total effect.

On page 535, Lieut. Schade says: "* * * lies in the proven fact that the model basin predictions of resistance and power are made with reasonable accuracy without taking account of the scoop resistance in the manner proposed." This is a question which I believe is open to considerable argument, since in those cases with which I am familiar where the thrust has been measured, neither the thrust nor the T.H.P. has been in agreement with the tow-rope resistance or the E.H.P. The difference between the measured thrust or T.H.P. and the tow-rope resistance or E.H.P., is explained by the wake and thrust deduction factors which are juggled in order to bring this into agreement. In the case of single-screw vessels of full form, long middle bodies and short runs, a considerable correction for thrust is undoubtedly justified. This does not apply, however, to high-speed ships with fine runs and the propellers operating mostly in substantially undisturbed water.

In the case of the destroyer discussed by Commander Saunders and Lieut.-Commander Pitre (U.S.S. *Hamilton*), the scoop correction brings the E.H.P. curve into much closer agreement with the T.H.P. curve and reduces the wake and thrust deduction factors to much more reasonable figures.

DISCUSSION OF MR. SCHMIDT'S REPLY.

By LIEUTENANT H. A. SCHADE (CC), U. S. NAVY.*

In the discussion of the effect of the energy remaining in the discharge water, Mr. Schmidt seems to be comparing the energy due to the *absolute* velocity of the water within the condenser with the energy calculated from the velocity of the discharge water *relative* to the ship, and concludes that there is a recovery of 1½ per cent as the result of such a calculation. A computation on such a basis seems incorrect. The discharge water has an absolute velocity made up of two components—the absolute velocity of the ship and the velocity of the discharge water relative to the ship. Furthermore, the energy in the discharge water based on its absolute velocity is the net kinetic energy imparted to the water by the scoop and condenser system, regardless of what happens within the ramifications of the system. The system takes water from rest (ignoring for the present the boundary layer) and after passing through the system the water has acquired a certain absolute velocity, and therefore a certain kinetic energy which is a function of this absolute velocity, and this kinetic energy is a measure of the power required, ignoring friction, internal energy and disturbance in the streamlines outside the ship which Mr. Schmidt proposes to cover by his f—factor.

If we let V_s be the velocity of the ship, and V_w the velocity of the discharge water relative to the ship, both in feet per second, the absolute velocity of the discharge water V is—

$$V = \sqrt{V_s^2 + V_w^2 - 2\,V_s\,V_w \cos\theta}$$

where Θ is the angle the discharge water makes with the ship's velocity. The power required is then

$$\text{S.H.P.} = \frac{1}{2}\,\frac{Q \times 8.6}{60\,g}\,\frac{f}{550\,E_p}\,[V_s^2 + V_w^2 - 2\,V_s\,V_w \cos\theta]$$

$$= \frac{Qf}{247{,}000\,E_p}\,[V_s^2 + V_w^2 - 2\,V_s\,V_w \cos\theta]$$

* Design and Construction, Bureau of Construction and Repair.

Using the values assumed by Mr. Schmidt for the 42-knot destroyer with common scoop

$Q = 80{,}000$ gallons per minute
$V_s = 71$ feet per second
$V_w = 12$ feet per second
$\Theta = 30$ degrees
$E_p = .53$
$f = 2.5$

the power required is

$$\text{S.H.P.} = \frac{80{,}000 \times 2.5}{247{,}000 \times .53}\left[(71)^2 + (12)^2 - 2\,(71)\,(12)\,(.866)\right]$$

$$= 5230 \text{ H.P.}$$

The percentage of power loss is

$$\frac{5230}{80{,}000} \times 100 = 6.5 \text{ per cent}$$

Comparing this with the 9.6 per cent loss calculated by Mr. Schmidt, it is seen that taking the discharge velocity into account reduces the loss by 32½ per cent, instead of 1½ per cent as stated by Mr. Schmidt.

Considering now the boundary layer effect, it is impossible to agree with Mr. Schmidt that this need not be taken into consideration. The condenser water is taken from a moving mass of water coming from ahead, not from outboard, and returned to the moving mass, receiving in the process additional kinetic energy. The added kinetic energy should therefore be evaluated on the basis of the moving boundary layer as the reference condition, not still water.

If we assume that the mean boundary layer velocity in the vicinity of the scoops and over their depth is 25 per cent of the ship's velocity (which is conservative, based on the *Hamilton* experiments) then the formula for S.H.P. loss becomes—

$$\text{S.H.P.} = \frac{Qf}{247{,}000\,E_p}\left[(.75\,V_s)^2 + V_w^2 - 2 \times .75\,V_s\,V_w \cos\theta\right]$$

and for the assumed conditions, this becomes

S.H.P. = 2870 or 3.6 per cent loss.

This figure compares with 9.6 per cent arrived at by Mr. Schmidt.

If now we do not apply the factor f, the theoretical loss is—

S.H.P. = 1148 or 1.4 per cent loss.

This figure compares well with Model Basin tests, which indicate that on a destroyer the difference in resistance between a model with dummy scoops and one with scoops where the water is allowed to pass through the hull is not over 2 per cent.

NOTES.

INDEX TO NOTES.

NAVAL CONSTRUCTION IN 1933.

By Hector C. Bywater, The Engineer, January 5, 1934.

THE VULCAN HYDRAULIC CLUTCH.

Motorship, December, 1933.

THE JOHNSON NAVAL TYPE BOILER.

The Engineer, December 29, 1933.

FULL SCALE TRIALS OF A DESTROYER.

Shipbuilding and Shipping Record, December 14, 1933.

VISIBILITY AT SEA.

Shipbuilding and Shipping Record, December 14, 1933.

THE PROGRESS IN THE DEVELOPMENT OF NAVAL GUNS.

Journal de la Marine Le Yacht, 30 December, 1933.

NAVAL CONSTRUCTION IN 1933.

By Hector C. Bywater, A. I. N. A.

Outstanding events of the year in the province of naval construction were the completion and trials of H.M. cruisers *Leander* and *Achilles*, the first ships of the new 7000-ton class; the introduction of a large building program by the United States, embracing thirty-two vessels, estimated to cost £55,000,000; and an Admiralty announcement to the effect that the cruiser program for 1933 had been altered from three ships of 5200 tons and one of 7000 tons to two ships of about 9000 tons and one of 5200 tons. The reasons for this important modification are set forth below. During the year work has continued on the French battle-cruiser *Dunkerque*, but at such a leisurely pace that her launching date remains indefinite. As the only true capital ship now building anywhere in the world, this vessel is naturally an object of interest. Germany launched her second miniature battleship on April 1st as the *Admiral Scheer*, and on the same day the prototype, *Deutschland*, was formally commissioned for service. With the passage of time the enthusiasm which this novel type aroused in technical circles has somewhat diminished, and the so-called "pocket battleships" are now regarded in their true light as an audacious experiment, the success of which has yet to be demonstrated. The silence which the German authorities have observed in regard to the *Deutschland's* trials may not be without significance, though the high reputation enjoyed by German shipbuilders and engineers, not least in connection with the development of marine oil engines of large power, renders it probable that the 56,800 B.H.P. Diesel plant installed in the *Deutschland* has fulfilled expectations. Sweden has launched the cruiser seaplane carrier *Gotland*, the first vessel of this type to be built. For the rest, the United States is now the only country in which 10,000-ton cruisers of the Washington Treaty type are still being constructed. Elsewhere the type has fallen into disrepute on account of its high cost and extreme vulnerability. The semi-official description of the latest U. S. 10,000-ton 8-inch gun ships as "armored cruisers" is clearly a misnomer. In a ship of this displacement, with engines of 107,000 H.P. and an armament of nine 8-inch 30-ton guns plus eight 5-inch anti-aircraft pieces, the margin of weight left over for protection must be very narrow. Such armor plate as is fitted must be comparatively light, and it is probably as true of these ships as of the British *County* group that one salvo of 8-inch shell, or a single torpedo or mine, would suffice to disable, if not to destroy.

BRITISH EMPIRE.

In the House of Commons on November 15th the First Lord of the Admiralty made a statement which may be summarized as follows: The 1933 program included one *Leander* class cruiser of 7250 tons and three *Arethusas* of 5400 tons each, armed respectively with eight and six 6-inch guns. "The policy of building cruisers of comparatively small tonnage had been adopted in the hope that other nations would follow our lead." It had been learned, however, that Japan had six cruisers of 8500 tons, each mounting fifteen 6-inch guns, building or projected, while the United States had legislated for four 10,000-ton cruisers, also to be armed with fifteen 6-inch. "If therefore, our program, already approved, were to be carried out, the new cruisers would be definitely inferior to those being developed by other Powers." As our total cruiser tonnage to be completed by 1936 is limited by the London Treaty to 91,000 tons, the Admiralty had been placed "on the horns of a very serious dilemma." They had to decide whether to go on building a larger number of inferior cruisers or to lay down fewer cruisers of heavier

tonnage and stronger armament. The second alternative had been chosen, and this meant that the 1933 program would be revised to include two new type cruisers of about 9000 tons, with increased armament, and one cruiser of the *Arethusa* type, of about 5200 tons. No increase in the cost of the program was involved by the change.

Naval opinion as a whole has approved the Admiralty's decision. To continue building ships which in battle could not face their foreign contemporaries would be a thoroughly bad policy. The glaring fact that British statesmanship at the London Conference put this country in such a position that it must either build second-class ships, or so few of the first-class as to leave the Navy desperately short of cruisers, invites comment which would be out of place in this article. It is, however, permissible to express the hope that at the next Naval Conference, due in 1935, the vital defense interests of the Empire will not be sacrificed to political expediency. There is, of course, much speculation as to the characteristics of the 9000-ton cruisers, which, unfortunately, are not to be laid down before next summer. They will, it is presumed, carry not less than twelve 6-inch guns, and if these are all to be mounted on the center line the adoption of triple turrets would appear unavoidable. The displacement should allow for reasonably strong armor protection.

H.M.S. *Leander*, the solitary cruiser of the 1929 Estimates, was commissioned at the end of March, four years after authorization. Her trials were protracted, but she is understood to be a very successful ship in all respects, able to attain her contract speed of 32½ knots without difficulty. The displacement is officially given as 7250 tons and the machinery develops 72,000 H.P. The ship has a very smart appearance at sea, the single large funnel, in conjunction with the superposed 6-inch turrets forward and aft, giving a remarkable impression of power. Of the three similar cruisers authorized in 1930, the *Achilles*—see Supplement—built by Cammel-Laird, was commisioned in October last. In the following month she made a noteworthy voyage from Gibraltar to Portland, covering the distance at an average slightly above 28½ knots at three-fifths full power. This, the best long-distance run at high speed hitherto recorded of any cruiser of corresponding tonnage, British or foreign, reflects great credit alike on the machinery, contractors and the engineering department of the ship. The *Orion* began her trials in October, but had to interrupt them owing to a slight defect in the turbines, which took ten days to adjust. The full-power and other steaming trials have since been completed satisfactorily. The third ship, *Neptune*, is still undergoing trials. The 1931 group of cruisers comprises the *Amphion* and *Ajax*, of the *Leander* class, and the *Arethusa*, the first of a new 5400-ton class, armed with six 6-inch guns. All three ships, the laying down of which was greatly delayed, will take the water this year, the *Arethusa*, building at Chatham, being the first to leave the ways. She should be followed by the *Ajax*, which Vickers, Ltd., are constructing at Barrow, and then by the *Amphion*, whose keel was laid at Portsmouth last June. Although very similar to the *Leander* in size, armament, etc., the wider spacing of their boiler-rooms will give the *Amphion* and *Ajax* two funnels apiece. The three cruisers of the 1932 program are the *Phaeton* (Swan, Hunter and Wigham Richardson) and *Apollo (Devonport)*, of the improved *Leander* class and the *Galatea* (Scott's Shipbuilding Company), which is a replica of the *Arethusa*. The two 9000-ton ships and the *Arethusa* unit of the 1933 program are all to be contract built and the orders will probably be placed in March.

The flotilla leader *Exmouth* and eight *E* class destroyers of the 1931 program are due for completion this year. Except for a slight increase in displacement, they are repetitions of the *A* class built in 1928, which set the pre-

vailing standard for British destroyers. The latest boats are of 1375 tons, with machinery of 34,000 H.P. for 35 knots, and an armament of four 4.7-inch guns and eight 21inch tubes. Probably no better type exists for all-round service, but just as we have been forced by foreign competition to increase the tonnage and armament of our cruisers, so we may eventually be compelled to build heavier destroyers to hold the balance against the formidable craft now being constructed abroad, and exemplified by the U. S. 1850-ton leaders and 1500-ton destroyers, mounting 5-inch guns; the French leaders of 2600 tons, with five 5.5-inch guns; and the Japanese destroyers of 1700 tons and six 5.1-inch guns. The flotilla leader *Faulknor* and eight *F* class destroyers were contracted for last May, presumably as "repeats" of the *E* boats. A landmark in submarine development was established during the year by the *Thames,* built under the 1929 Estimates by Vickers-Armstrongs. Inaugurating the *River* class, this vessel has a surface displacement of 1805 tons, the length being 345 feet, and the beam 28 feet. Her oil engines, built to a design jointly worked out by the Admiralty and Vickers, are of 10,000 B.H.P., for a contract speed of 21¾ knots. On her trials, however, the *Thames* reached 22½ knots, a velocity never before attained by a submarine propelled by I.C. engines. The *Severn* and *Clyde,* of the same class, are also building at Barrow. The submarine minelayer *Porpoise,* of 1500 tons, was completed during the year at Chatham, where her sister boat *Grampus* is on the stocks. Of the medium coastal type, *Swordfish* class, seven boats have been completed and laid down to date, and two more are projected. Displacing only 640 tons, these boats are too small for overseas duty, but within their limitations they should prove useful additions to the fleet. The speed of 13¾ knots seems inadequate, but the armament of six tubes concentrated in the bows is formidable. A great advantage of the type is its low cost compared with that of the 1450-ton *Overseas* class which preceded it.

The sloops *Grimsby* and *Leith* have been launched at Devonport. Slightly heavier than the *Sandwich* class, sixteen of which have been built, their tonnage is 1060 as against 1025; the speed remains at 16 knots, but two 4.7-inch guns have been mounted instead of 4-inch. Under the 1931-32 programs four sloop minesweepers are in hand—*Halcyon, Skipjack, Harrier,* and *Hussar.* They are craft of 875 tons and 16 knots speed, mounting two 4-inch guns. The *Bittern,* convoy sloop, and *Kingfisher,* coastal sloop, both of the 1933 program, were ordered in the autumn. No details are available, but the *Bittern* is understood to be a larger vessel than any sloop we have built since the war. Having regard to the acute shortage of cruisers, it is important that the sloops which deputize for them should be reasonably large and powerful vessels. Unfortunately, that cannot be said of the sloops now in service or building. Doubtless for reasons of economy, their dimensions are too small to permit of the mounting of a really effective armament, and the only wartime duty they could perform efficiently is that of sweeping mines. While it is true that the sloop can never replace the cruiser proper, there is no good reason why it should not be made capable of undertaking the vitally important task of convoy protection, a duty for which our present sloops are quite unsuitable. At a comparatively small extra cost we could build sloops which would not only uphold in time of peace the prestige of the flag more worthily, but in war would afford real protection to ocean convoys, even against heavily armed submarines and surface corsairs of the *Moewe* type. And an unlimited number of such vessels might be built without any infraction of treaty obligations. Looking abroad, we see such sloops as the French *Bougainville,* a ship of 1968 tons, a speed of 15½ knots, and an armament of three 5.5-inch guns, and the recently completed Netherlands vessel *J. V. van Nassau,* of 1520 tons, 15 knots, and three 5.9-inch guns. It is certainly not beyond the capacity of our naval architects to design a sloop

of less than 2000 tons, with a speed sufficient for convoy work, say 16-17 knots; a battery of three 6-inch guns, a cruising radius of 5000 miles, and some armor protection. Half a dozen of these vessels could be built for the price of a 7000-ton cruiser. Useful enough in peace, they would be invaluable in an emergency. The *Bittern*, now on order with John Brown and Co., Ltd., may prove to be the forerunner of such a class.

UNITED STATES.

Last June President Roosevelt authorized a building program as part of his national industrial recovery scheme. It comprises thirty-two vessels, and is estimated to cost about £55,000,000. The program is to be completed within three years, and all contracts had been awarded before the end of August. American shipyards are therefore facing a period of prosperity. As the new program overlaps a considerable amount of tonnage authorized previously, it will be more convenient to deal with the United States construction now in hand as a whole. On January 1st, 1934, the following vessels were on the stocks or completing afloat: Three aircraft carriers, *Ranger* (13,800 tons), *Yorktown, Enterprise* (each 20,000 tons); seven 10,000-ton cruisers with 8-inch guns, *New Orleans, Astoria, Minneapolis, Tuscaloosa, San Francisco, Quincy, Vincennes;* four 10,000-ton cruisers with 6-inch guns, *Savannah, Nashville, Brooklyn, Philadelphia;* eight flotilla leaders of 1850 tons; twenty-four destroyers of 1500 tons; six submarines; two gunboats. Of these fifty-four vessels, the following are building by contract: Three aircraft carriers, three 8-inch cruisers, two 6-inch cruisers, sixteen leaders and destroyers, and three submarines. The remaining vessels have been allotted to the Navy Yards at New York, Philadelphia, Boston, Norfolk, Mare Island, and Puget Sound. Mr. Swanson, Secretary for the Navy, stated in November that over a hundred additional vessels must be built to bring the United States Navy up to treaty strength. Whether or no those extra ships are to be built, the current program is evidence of national determination to create and maintain a fleet of the first rank. To this policy no exception can justly be taken abroad.

The *New Orleans,* to be commissioned this month (January), is the eleventh U. S. cruiser of the 10,000-ton, 8-inch gun type to be completed. She differs from her predecessors in having the forecastle deck brought well aft, the bow has no pronounced overhang, and pole masts have taken the place of tripods. A belt of thin armor is reported to extend over two-thirds of the water line, weight having been saved by the use of electrical welding wherever possible and the adoption of a lighter type of 8-inch gun. Despite these improvements, however; the *New Orleans* perpetuates in large measure the inherent faults of her type, presenting as she does a large and inadequately protected target to gunfire and torpedo attack. Four further units of this class—*Astoria, Minneapolis, Tuscaloosa,* and *San Francisco*—are due for completion in the present year. The *Quincy* was begun in 1933, and the keel of the *Vincennes* was laid on January 1st, 1934. This will leave only one more ship to complete the American quota of eighteen such vessels, as against thirteen British and eight Japanese.

Of greater interest, perhaps, are the four 10,000-ton "light cruisers," of the *Savannah* class, building under the new program. Their design was the subject of representations made at Washington in September by the British Government, which drew attention to the fact that the building of such large vessels was liable to stimulate competition. Great Britain has all along been anxious to restrict future cruisers to 7000 tons, which she regards as adequate to fulfill all the duties of their type. Beyond exciting some resentment in the American press, this *démarche* appears to have had no effect. The *Savannah*

and her sisters are being built to the original design, and as 8-inch guns are disallowed by treaty, they are to have the imposing armament of fifteen 6-inch guns, presumably mounted in triple turrets. There is an unconfirmed report that one of the group may have a partial flight deck for aircraft, her armament being reduced to nine 6-inch. So far, however, no official details have been disclosed.

The aircraft-carrier *Ranger* is to be delivered on May 1st, her launch having taken place on February 25th, 1933. The length is 727 feet on the water line, the beam 80 feet, draught 19 feet, and the displacement 13,800 tons. The ship is propelled by geared turbines of 53,500 S.H.P., but her contract speed is not yet public. She has hangar stowage for seventy aircraft. The original design provided for outboard smoke-ducts, but it has since been decided to fit a funnel and an "island" superstructure on the extreme starboard side of the flight deck. The armament consists of eight 5-inch A.A. guns. The *Ranger* is interesting as an example of the medium-size carrier of which H.M. ships *Argus* and *Hermes* and the Japanese *Hosho* and *Ryujo* are earlier specimens. The new U. S. carriers *Yorktown* and *Enterprise*, now on the stocks at Newport News, are much larger ships, the displacement being 20,000 tons. Beyond this no particulars have been released.

Eight flotilla leaders building under the new program will be the first vesesls of this type in the U. S. Navy. Conforming to the maximum size permitted by treaty they displace 1850 tons; the contract speed is 37 knots, and the armament includes six 5-inch guns and eight tubes. The twenty-four destroyers appear to be uniform in design: 1500 tons, 37 knots, five 5-inch, 40-caliber guns, eight tubes. In all these vessels the cruising endurance is exceptionally high, 6000 miles at economical speed in the destroyers and probably more in the leaders. They compare well with the big Japanese destroyers of the *Fubuki* class, to which, in some measure, they probably constitute a reply. All six submarines now building are boats of about 1300 tons, the construction of submarine cruisers, such as the *Argonaut, Narwhal,* and *Nautilus*—2700 tons, two 6-inch guns—having been suspended.

Except in so far as the new 6-inch gun cruisers appear to British eyes, needlessly large, the new U. S. program is a well-balanced piece of legislation, framed on sound technical lines dictated by purely service requirements and unhampered by the political considerations which have restricted both the number and fighting power of recent British warships.

No review of U. S. naval building would be complete without some reference to the large scheme of battleship modernization which is now within sight of completion. On each of the last five ships to be dealt with a sum of £3,000,000 is to be spent, bringing the total bill for renovating fifteen battleships to approximately £38,000,000, equivalent to the cost of five *Nelsons.* The expenditure of this enormous sum indicates the value which American naval opinion attaches to a well-found battle fleet. In every case reconstruction has involved the fitting of bulges and additional armor, and in some instances an increase in gun elevation and an entirely new machinery outfit. In the *New Mexico,* for example, the original turbo-electric drive is being replaced by geared turbines. As a result of this thorough-going modernization the U. S. battle fleet must now be rated well above its paper strength, and may, indeed, be considered the most formidable in existence.

JAPAN.

No large ship went afloat during the year, and the only one to be completed was the aircraft carrier *Ryujo.* This vessel, launched at Yokohama in April, 1931, has the following characteristics: Length, 548 feet; beam, 60 feet 6 inches; draught, 15 feet 4 inches; displacement, 7600 tons. She is

driven by geared turbines developing 40,000 S.H.P., her speed being 25 knots, and is armed with twelve 5.1-inch A.A. guns on twin mountings. Her aircraft capacity has not been divulged. In appearance she is a miniature *Akagi,* smoke being ejected from horizontal ducks. The cruisers *Mikuma* and *Mogami,* the first two of a group of six, will be launched shortly. In common with all recent Japanese cruisers they embody novel features. It is probably not too much to assert that these ships are directly responsible for the design of the American 10,000-ton, 6-inch gun cruisers, and indirectly so for that of the new British 9000-ton class. The following details of the *Mogami* are understood to be official: Length, 625 feet; beam, 59 feet 6 inches; draught, 14 feet 8 inches; displacement, 8500 tons. Machinery, 90,000 S.H.P. for 33 knots. The armament, originally returned as fifteen 5.1-inch, is now known to consist of that number of 6.1-inch guns disposed on the center line in triple turrets. On paper, at least, the ship appears to be over-gunned, and if the engine power is accurately stated, there can be little or no weight to spare for armor protection. Japanese designers have in the past displayed much ingenuity in compressing great fighting power into moderate displacement, and it may, therefore, be prudent to withhold criticism of the *Mogami* and her consorts until their performance is known.

All twenty-four "super-destroyers" of the *Fubuki* class have been launched. As they were fully described in last year's review, it need only be said that they are ships of 1700 tons and 35 knots, armed with six 5.1-inch guns and nine 21-inch tubes, and notable in having the guns paired in closed turrets and splinter-proof hoods over the torpedo tubes. I am informed by a Japanese naval officer that this unusual arrangement was adopted after a searching investigation by the Japanese naval staff of destroyer tactics in the Great War. Apparently, it is intended to prevent the summary disablement of gun crews and torpedo parties by shell splinters. At the other extreme, from the viewpoint of size, are the new torpedo-boats of the *Chidori* class, four of which are completing afloat. They are boats of 527 tons, with a speed of 26 knots, and are armed with three 5.1-inch guns and four tubes. It would not be unreasonable to describe them as lineal descendants of the British *P* boats of 1917, which were intended primarily to hunt submarines. In view of the heavy weather often encountered in Japanese waters, the tactical value of these little craft must needs be limited. Five submarines, of 1975 to 1430 tons, are under construction. A large "replenishment" program is now under consideration, but has not been adopted at the time of writing. New construction already in hand or shortly to be started embraces, apart from cruisers, twelve 1400-ton destroyers, six submarines, one large and three small minelayers, and various nondescript vessels.

FRANCE.

Some further details of the *Dunkerque* have been published in the French papers. Although this vessel was begun at Brest in December, 1932, she is reported to be making such leisurely progress that the launching date cannot yet be fixed. The official dimensions are: Length, 689 feet; beam, 98.4 feet; displacement, 26,500 tons. Machinery: geared turbines of 100,000 S.H.P.; contract speed, 29½ knots. The main armament of eight 13-inch (33-centimeter) guns is grouped forward in two quadruple turrets, each pair of guns in the turret forming an autonomous unit, divided from the other pair by a massive bulkhead. In effect, therefore, the quadruple gun-house represents twin turrets mounted on a common turntable. All eight guns have a wide arc of training abaft the beam. Sixteen 5.1-inch (13-centimeter) guns constitute the secondary armament, mounted in two double and three quadruple turrets. The main armor belt, which begins at the base of the fore-

most turret and is carried aft to within about 150 feet of the stern, is 10.8 inches on the water line. Above this belt there is an armor deck 4.9 inches thick, and below it a second deck of 2-inch plating. Armored longitudinal bulkheads are worked over the greater part of the ship. The total weight of armor is said to be 10,000 tons. If this be true the percentage of protection to displacement is higher than in any previous ship of war. It is clear from these revised details that the *Dunkerque* will be much more formidable than the first reports indicated. Although officially rated as a *cuirassé*, or battleship, her attributes are those of a battle-cruiser second in fighting power only to H.M.S. *Hood*.

The *Algérie*, completed at Brest in the autumn of 1933, is the seventh, and probably the last, of the French 10,000-ton *Washington* cruisers. Although the speed has been reduced to 32 knots, she is patently superior in military value to her predecessors. The weight saved in machinery has been put into protection, consisting of a broad 4-inch belt over some three-fifths of the water line and spanned by an armor deck; further, the subdivision to localize torpedo damage has been improved. In place of a foremast the *Algérie* has a large control tower, which appears to be a modified version of H.M.S. *Nelson's* famous structure. *La Galissonnière*, first of a class of six cruisers authorized in 1930-32, was launched at Brest on November 17th. The dimensions, etc., are: Length, 590.5 feet; beam, 57.4 feet; draught, 16.4 feet; displacement, 7729 tons. Machinery: geared turbines of 84,000 S.H.P. for 31 knots. As in the case of the *Algérie*, a reduction in speed has been accepted for the sake of improved armor protection, which is designed to resist 6-inch shell fire. The main armament is nine 6.1-inch guns in triple turrets, two forward and one aft; eight 3.5-inch A.A. guns, and four 21.6-inch torpedo tubes. Two seaplanes will be carried. The design of this ship is eminently workmanlike and thorough, a remark that applies equally to almost all the post-war vessels of the French Navy. A sister ship, *Jean de Vienne*, is shortly to be launched at L'Orient, and the *Gloire, Marseillaise, Montcalm*, and *Georges Leygues* are building in private yards. France has done well to perpetuate in the last-named cruiser the memory of her great Minister of Marine, who died last year, since to him, far more than to any other individual, she owes the rebirth of her national sea power.

The cruiser minelayer *Emile Bertin* went afloat at St. Nazaire on May 9th. If she comes up to expectations she will be one of the most remarkable ships of the day. This is what her designers have put into a hull 548 feet long and 52 feet 6 inches broad, drawing 16.4 feet, on a displacement of 5886 tons: Machinery of 102,000 S.H.P. for a speed of 34 knots; an armament of nine 6.1-inch guns in triple turrets, plus twelve light A.A. guns and six torpedo tubes, and a large outfit of mines with laying gear. And in addition to all this an electrically welded armor deck has been worked over the vitals of the ship and provision made for a catapult and seaplane. If the steam and gunnery trials of this ship are successful she will represent a new triumph for French naval architecture.

Several flotilla leaders of the belated 1930 program took the water during the year. Officially listed as *contre-torpilleurs*, these vessels are cruisers in all but name, and are rated as such by the London Treaty. *Le Malin*, launched in August, and *L'Indomptable*, in November, belong to the new two-funnelled class, in which the displacement has been increased from 2440 tons to 2570 and the S.H.P. output from 70,000 to 75,000, raising the speed from 36 to 37 knots. The gun armament is the same as in the preceding class, viz., five 5.5-inch, but nine torpedo tubes are carried instead of seven. Thirty-one of these exceptionally powerful ships have been laid down to date, the smallest being the six *Chacals*, of 2126 tons and 35½ knots. All have exceeded their contract speeds on trial, in some cases by as much as five or six knots. The

"record" is held at present by the *Cassard,* which reached 42.9 knots, while the *Gerfaut, Epervier,* and *Milan* have all steamed at more than 42. It is worthy of note that these vessels do not serve as flotilla leaders, but are organized in divisions. This suggests that they are intended to be used as light cruisers rather than as torpedo craft. The only destroyer proper now building in France is *Le Hardi,* an experimental boat credited with a displacement of 1600 tons. A group of twelve torpedo boats, officially designated *escorteurs* (convoy guards) is under construction in private yards. The displacement is 600 tons, the length 236.2 feet, the contract speed being 35 knots. Two 3.5-inch and some smaller guns, with four 15.7-inch torpedo tubes, comprise the armament. These craft should be useful for patrol and anti-submarine duties, and they are of interest as marking a return to true torpedo boat design. In spite of delays in the building program the French submarine force continues steadily to grow in strength. Thirty boats of the well-tried *Redoubtable* class are now completed or building, hardly any modification having been made in the original design: 1644 to 1678 tons surface displacement, 18 knots, one 3.9-inch gun and eleven tubes. Many of the boats have made long voyages and are capable of remaining at sea for a month. They are larger editions of the *Requin* class (nine units), and appear to be thoroughly successful craft. A large number of coastal boats, ranging from 558 to 571 tons, are still in hand, and eventually thirty-one will be completed. Excluding over-age boats, the French Navy now has eighty modern submarines ready or constructing, a total exceeding that of any other navy.

ITALY.

The *Bolzano*—see Supplement—the seventh 10,000-ton *Washington* cruiser, was completed by Ansaldo during the year. In this ship there has been a return to the very high speed, 35.5 knots, achieved in the *Trieste* and *Trento,* at the sacrifice of that substantial armor protection which is a feature of the intervening 32-knot vessels, *Zara, Fiume, Gorizia,* and *Pola.* Now that Italy has gained numerical equality with France in this type of ship, it is unlikely that she will build any more. Her naval architects are concentrating on the development of the *Condottieri* class of light cruisers, twelve of which have been laid down since 1928. There has been a progressive increase in displacement, from 5070 tons to 7000, with a corresponding advance in S.H.P. to maintain a uniform speed of 37 knots. The armament remains unchanged at eight 6-inch guns. On trial, the earlier ships have steamed at 39 to 40 knots, thus establishing a new record for cruisers. Besides their other qualities, they are exceptionally handsome ships, bearing all the marks of bold yet careful design. Four destroyers of 1449 tons, with a contract speed of 38 knots, are in hand. They will complete a fairly homogeneous flotilla of twelve boats distinguished by their large single funnels. Since completing the twelve 1628-ton vessels of the *Navigatori* class three years ago, Italy has made no further attempt to reply to the huge French leaders. Eight submarines, the largest a minelayer of 1393 tons, are building.

OTHER NAVIES.

The Swedish "aircraft cruiser" *Gotland* was launched on September 14th. She was briefly described in last year's review, but the novelty of her design merits a further short notice. She is 426 feet 6 inches in length, 47 feet 9 inches in beam, and draws 16 feet on a displacement of 4600 tons. The machinery consists of two sets of de Laval geared turbines taking steam from four small tube boilers and developing 33,000 S.H.P. for 27 knots. A considerable amount of light armor has been worked in to protect the waterline,

boiler uptakes, and gun positions. Six 6-inch guns are mounted, four in double turrets at bow and stern, and two in single turrets on the beam abreast of the bridge. The after part of the ship, representing one-third of the total length, is reserved for a catapult and seven or eight seaplanes. Much ingenuity has been displayed in the arrangement of armament and aircraft equipment. A vessel of this type would be a valuable adjunct to any fleet, and it will not be surprising if the basic design reappears in modified forms in other navies.

After trials lasting five months the German *Panzerschiff Deutschland* was commissioned on April 1st. No official details of the trials have been released. On the same day a sister ship, *Admiral Scheer,* was launched at Wilhelmshaven. A third unit, *Ersatz Braunschweig,* is to be launched this spring, and a fourth *Ersatz Elsass,* will be laid down at an early date this year. The class has been exhaustively described in previous annual reviews, and there is little new to report beyond an increase in the torpedo armament. This now consists of eight tubes on quadruple carriages, one at each side of the quarterdeck, each mounting fitted with an armored hood to protect it from the blast of the after 11-inch turret.

British industry is playing a prominent part in the rebuilding of the navy of Portugal, our oldest ally. Two sloops—see Supplement—which almost deserve promotion to the cruiser category, are building by Hawthorn, Leslie. They are trim little ships of 2100 tons, with turbine machinery of 8000 S.H.P. = 21 knots, and an armament of four 4.7-inch guns, two 3-inch A.A. guns, and accommodations for forty mines. The same firm delivered to the Portuguese Government last year two smaller sloops, and Yarrows completed the destroyers *Vouga* and *Lima,* which closely resemble the latest British type. Vickers-Armstrongs are completing three Portuguese submarines of 854 tons surface displacement. These contracts are another indication of the tendency of foreign Governments again to make use of the unrivalled resources of the British shipbuilding and armament industry when it is a question of augmenting their naval forces with material of the highest class. —"The Engineer," Jan. 5, 1934.

THE VULCAN HYDRAULIC CLUTCH.

As the result of broader and more diverse marine applications of the Diesel engine, new trends in engine design become apparent and the most noticeable of these is the movement toward higher speed, lighter weight, smaller dimensions and greater power in a single unit. Progress along these lines has already reached a point where crankshaft speed of the prime mover is much higher than the most desirable propeller speed in commercial ships, naval vessels and even in certain types of pleasure craft.

This shifting into the higher speed realm has become so general as to almost definitely preclude any question of its advisability, while in many instances it proved to be the only means of securing the advantageous features of the Diesel. Conspicuous amongst the instances of high-speed Diesel application are the inroads which the Diesel is making in naval work, other than submarines, which already owe their success to the earlier achievements in high-speed, light-weight oil engine development.

It goes without saying that any difference between the prevailing rates of speed of modern Diesels and efficient propeller speed must result in the use of a speed reducing element in the propulsion system. Two leading methods have been tried and proven in service. One of these is Diesel electric drive which has been the subject of considerable discussion, and Diesel gear drive

which is extensively employed but has received less general notice, although actual installations of gears have been made in connection with the largest Diesel power plants afloat. It is now evident that new and larger Diesel power applications will be made in marine service through the use of gears, especially where multiple power units are applied to a single propeller shaft. It is also evident that, where multiple power units are so employed it is necessary to provide a means of dropping out or engaging one of the power units without interfering with the others. This has been accomplished by means of Vulcan clutches of the hydraulic type produced by the Hydraulic Coupling Corp. of Detroit, Michigan, developed by the Vulcan Werke of Hamburg, Germany, and the Hydraulic Coupling Patents, Ltd., of England.

This type of clutch has certain inherent advantages, notably its ability to transmit power without transmitting torsional vibration or torque fluctuations —in short, the clutch transforms uneven torque of the reciprocating engine to almost completely even torque in the line shaft. This is due to the fact that there is no solid connection between the driving and driven elements, the power of the former being transmitted to the latter by an hydraulic bridge across a fixed gap which separates two halves of the clutch. With this type of clutch it is possible to prevent the operating characteristics of one power unit from interfering with another unit while both supply power to a common service.

As the load can be dropped or picked up instantly (within 3 seconds) without shock there is no need of synchronizing the various power units either as to load or speed and there will be no undue stress developed in the process of engaging the clutch. And, while it is quite evident that these various characteristics are particularly desirable where multiple power units are employed, it is also apparent that the clutch can be employed advantageously where there is only one power unit that must be periodically relieved of the line shaft load as, for example, in the submarine which is driven alternately by a Diesel and an electric motor.

Installations have been made in different types of vessels from the 56,000 horsepower twin screw Diesel battle cruiser *Deutschland* with four engines per shaft line to several 600 horsepower sternwheel Diesel tugs operating in Columbia, S. A. Noteworthy passenger vessels so equipped include the 12,000 horsepower installation in the twin screw 18,000 ton motorship *St. Louis* while in cargo ships the 5200 horsepower installation on one shaft in the *Kota Agoeng* is worthy of note. In the smaller range we find such motorships as the 6300 ton *Wulsty Castle* with 1600 horsepower on a single shaft; the *Rendsburg* with 4100 horsepower on a single shaft and a wide variety of other types of craft including a 3000 horsepower single screw tug with two engines for a Dutch firm and a 300 horsepower dredge in the Dutch East Indies.

The foregoing examples are sufficient to indicate the breadth of power range to which the clutch has been applied and extensive variations in motor vessel types in which it has been employed.

Design of the clutch is interesting but it is capable of being mistakenly analyzed. Following the first reference to the Vulcan clutch and gear which appeared in *Motorship* and from time to time thereafter we have received inquiries regarding its operation and the services in which it can be employed. The Vulcan hydraulic clutch is designed for marine service and one of its functions is to engage and release a power unit, differences in speed of the engine and propeller being definitely established and maintained by reduction gears. There is another design of this clutch, the Vulcan Sinclair coupling which affords infinite variable speed regulation and is used in various industrial services. In the marine unit there is a slippage that is controllable to a

certain extent by the fluid medium, generally lubricating oil, employed to transmit power from one-half of the clutch to the other. To understand this it is necessary to examine the constructural plan which we have reproduced in Figure 2.

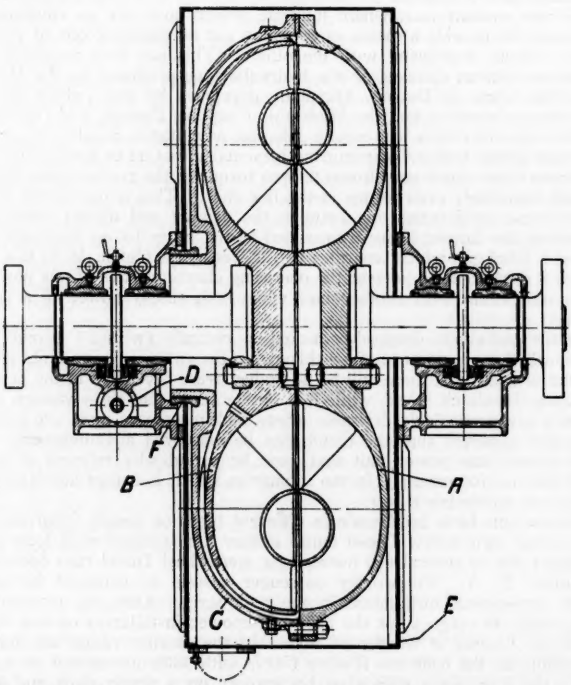


FIGURE 2.—SECTIONAL ARRANGEMENT OF AN HYDRAULIC COUPLING.

In the line drawing illustration is shown a driving wheel at the right and a driven wheel at the left. These parts are designated as A and B respectively in the line drawing. The wheels are flanged to the driving shaft (engine crankshaft) and the driven shaft which carries the pinion in the reduction gears. The assembled units are mounted in their respective bearings so that there is a clearance of about ¼ to ½ inch between the faces of the two wheels. It will be noted that the driving wheel (A) carries a shroud which extends over and completely encloses the driven wheel (B). This shroud is bolted fast to the driving wheel. It will also be noted that radial fins* are arranged at regular intervals in each wheel. In service, the entire assembly is covered with an oil tight housing.

* Figures 3 and 4 taken from the Vulcan-Gearing and Hydraulic Couplings catalogue are included in order to more clearly illustrate the construction and operation of the clutch.

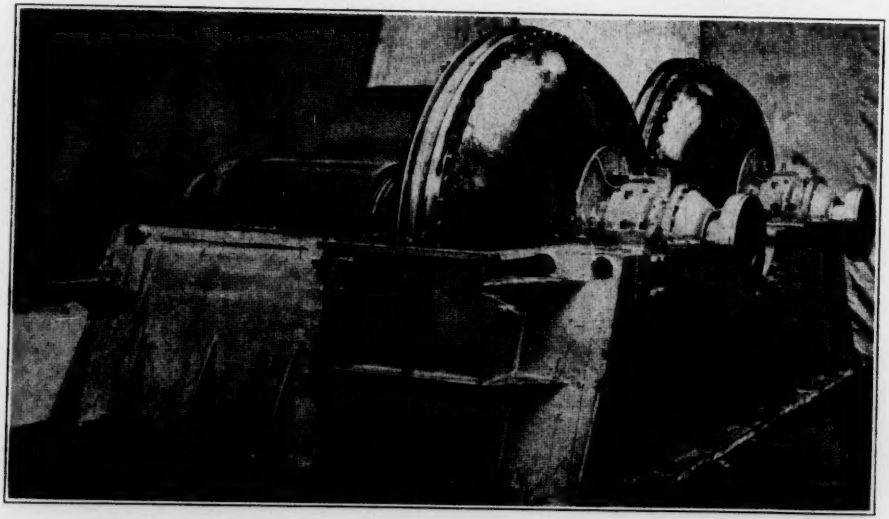


FIGURE 1.—HYDRAULIC CLUTCHES AND REDUCTION GEARS ASSEMBLED, WITH COVER REMOVED.

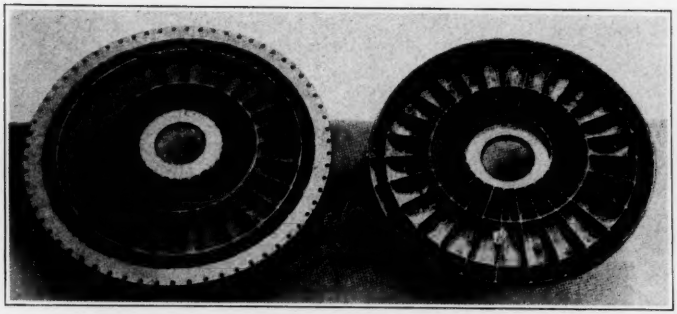


FIGURE 3.—PRIMARY AND SECONDARY WHEELS OF AN HYDRAULIC COUPLING.

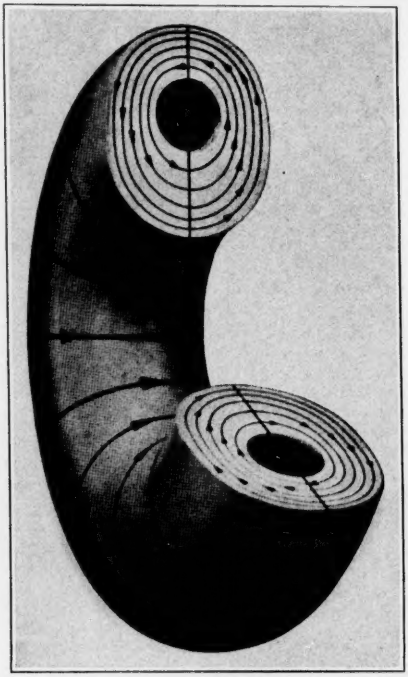


FIGURE 4.—MODEL SHOWING THE MOVEMENT OF THE FLUID INSIDE AN HYDRAULIC COUPLING.

While the coupling is empty there would be no movement of the secondary wheel due to revolving motion of the primary wheel, there being no positive connection between them, but when the space between the two wheels is filled with liquid such as lubricating oil the secondary wheel also revolves. Oil for this purpose is admitted through a passage in or beside the shaft. Centrifugal force caused by the revolving wheel throws the oil to the outer extremity of the circle at which point it crosses into the secondary wheel. As the oil flows across the gap between the two wheels it is caught by the radial fins in the secondary unit and power is transmitted from one wheel to the other by the impulse of the moving mass of oil. In actual service the oil flows from the primary to the secondary wheel at the outer rim of the circle and flows back from the secondary to the primary wheel along the inner rim of the circle.

Although the driving wheel always revolves somewhat faster than the driven wheel, the difference in speed of the two halves of the unit diminishes as all of the air is displaced by oil, at which point the difference in speed is about three per cent. As previously noted, the driving wheel is shrouded completely enclosing the driven wheel and this shroud keeps the inner parts of both halves of the unit completely immersed in oil. The loss in efficiency is directly proportional to the difference in speed so that the average unit of this type has a mechanical efficiency of about 97 per cent which has been found to be the best commercial compromise. It should be noted that every clutch of this type can be designed for a slip of about 1 per cent and 99 per cent efficiency if sufficient space is available.

The power lost in the clutch is due to fluid friction which is transformed into heat. In order to avoid excessive temperature within the unit, small holes are arranged in the circumference to allow a certain amount of the driving fluid to escape into the outer housing from which it is drained and pumped back into the clutch, if necessary through a cooler. The fluid friction between the primary wheel and the shroud on the secondary wheel is very low as the difference in speed between the two is small.

To disengage the clutch it is merely necessary to stop the supply and clear the unit of fluid as quickly as possible. A number of methods may be employed and it may be assumed that the most convenient arrangement for a particular installation will be chosen. A method which is being used with entire success in the large marine installations is to provide a series of holes in the outer rim of the primary wheel and cover these holes with a ring which is moved laterally an amount sufficient to uncover the holes whereupon the centrifugal force of the revolving unit throws out all of the fluid in a matter of two or three seconds. The time required to re-engage the clutch will be entirely dependent upon the capacity of the pump employed in refilling the clutch and maintaining the circulation required in service.

In describing the shroud on the primary wheel and how it holds the fluid in place we answer a question which has been frequently asked as to how the oil is held in place and how the design compensates for what would seem to be a tendency of the two halves to separate. Suitable thrust bearing provision is made to take care of any inclination to change the running clearance between the two wheels.

It is quite apparent that extensive variations in design of the clutch can be employed to meet different conditions of operation. A splendid example is its use in foreign submarines where the space premium is high and this condition has been met by a double unit wherein two primary wheels are attached to the driving shaft and two secondary wheels to the driven shaft resulting in a diameter reduction of some 15 per cent. Another noteworthy instance is the incorporation of two Vulcan clutches in the combined speed reducing and reversing gear on the Hamburg-American Line motorship *Heidelberg*. This

gear was especially designed and built to meet the requirements of marine propulsion with non-reversing Diesels.

In this country, interest in the Vulcan hydraulic clutch for use in connection with Diesel-gear drive seems likely to be more pronounced than in the past. It is well known that our navy is moving toward the adoption of Diesel propulsion for surface craft. As a prelude to more extensive work along these lines we may point to the conversion of several navy tankers wherein the existing shaft lines will be retained and high-speed Diesels will be employed in multiple units with suitable provision being made to bridge the speed gap between the prime movers and the propellers.

In point of weight and space saved the German naval work with Diesels provides the most interesting study and that work has been along the lines of reduction gear drive with Vulcan clutches. Since the experience already gained in this work abroad is available through the American manufacturer of the Vulcan clutch and since reduction gears for marine use have already been highly developed in this country, it is interesting to speculate on the possibility of it playing an important part in our navy power plants of the future.

It is of equal interest to contemplate ultimate influence on our merchant ship power plants in the event of carrying out any such plan. It is not beyond the bounds of reason to predict that pioneering work done by the navy along these lines will be even more closely imitated by the merchant marine than has been the system of electric propulsion which gained its present position of prominence through naval acceptance.—"Motorship," Dec., 1933.

THE JOHNSON NAVAL TYPE BOILER.

Successful tests have recently been completed at the works of John Samuel White and Co., Ltd., East Cowes, on a "Johnson" marine type water-tube boiler, manufactured by Clarke, Chapman and Co., Ltd., of Gateshead-on-Tyne, for demonstration to the Admiralty. We recently attended full-power and over-load trials of this boiler, of which a description is herewith given.

The Gateshead firm, it will be recalled, now holds the British and foreign rights to manufacture and to grant sub-licenses for the manufacture of this boiler. The first unit of this type incorporating an air heater and superheater was built by John Brown and Co., Ltd., at Clydebank, successful tests being completed in January, 1929. The maximum rate of steam generation was 13 pounds per square foot of heating surface. It was subsequently installed in the Canadian Pacific liner the *Empress of Britain,* as one of a battery of nine main boilers, and has been in regular use ever since that ship was commissioned in May, 1931, working at a rating of 10 pounds per square foot of heating surface, or double that of another type of boiler in the same installation.

A smaller boiler, having an air heater, but no superheater, was built by another firm in 1930, and installed in the Canadian Pacific Railway Company's steamer *Princess Helene,* in conjunction with two boilers of a well-known design. Prior to the official trials, the "Johnson" boiler was tested up to a rating of 12 pounds per square foot of heating surface, corresponding to 14.7 pounds from and at 212 degrees F. It was utilized in conjunction with the other two main boilers throughout the official speed and power trials, on the completion of which the vessel steamed from Greenock to St. John, N. B., at a speed of 14 knots with only the "Johnson" boiler in use for propulsion.

The " Johnson" boilers fitted in these two ships were the outcome of efforts on the part of the designer to produce a water-tube boiler suitable for merchant vessels, and capable of being worked continuously at a rating of at least 10 pounds, with high efficiency. The results obtained in the two vessels mentioned have been so satisfactory that, after due investigation by the builders' and owners' representatives, it was decided to adopt this type of boiler for the Royal Mail liners *Asturias* and *Alcantara*, in which the space available is unusually restricted for the power required. These boilers are now in an advanced state of construction.

The British Admiralty having exhibited interest in the potentialities of the boiler for naval purposes, a design was prepared and a unit duly constructed for demonstration by Clarke, Chapman and Co., Ltd. The results of the tests on this boiler constitute a notable technical achievement in the sphere of naval boiler performance. The essential data are given below, permission for publication having been granted by Vice-Admiral Brown, Engineer-in-Chief of the Fleet. The boiler, which is shown in Figures 1 and 2, has a generating surface of 9020 square feet and superheater surface of 1810 square feet, and

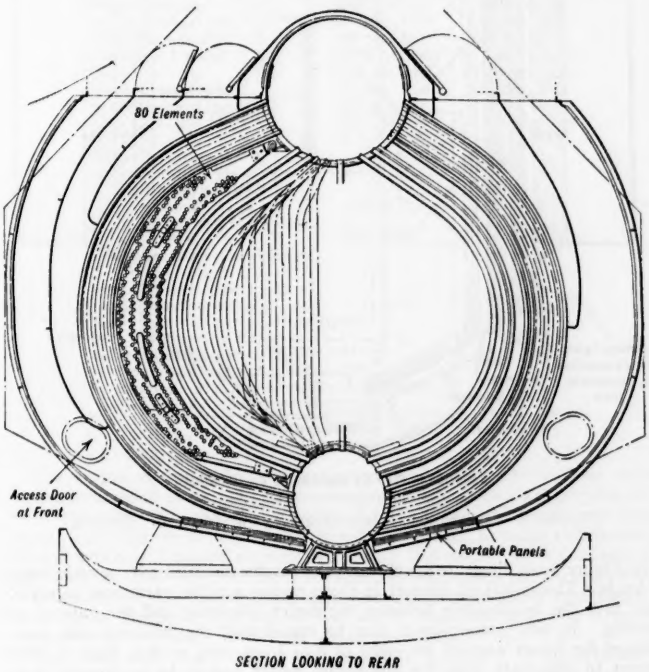


FIGURE 1.—SECTIONAL ARRANGEMENTS OF JOHNSON BOILER.

was designed for a continuous evaporation of 165,000 pounds per hour. This output was, in fact, maintained during a special test for a period of twenty-four hours. The specified air pressure corresponding to this output was 4 inches, but with all ten burners on, it was actually 3.8 inches. An overload test of four hours' duration at a mean evaporation of 192,000 pounds was conducted later, and subsequent examination of the fire tubes and refractory revealed no signs of distress or damage. In both these tests exceedingly good combustion was obtained with standard Admiralty burners. The boiler was not provided with an air heater:

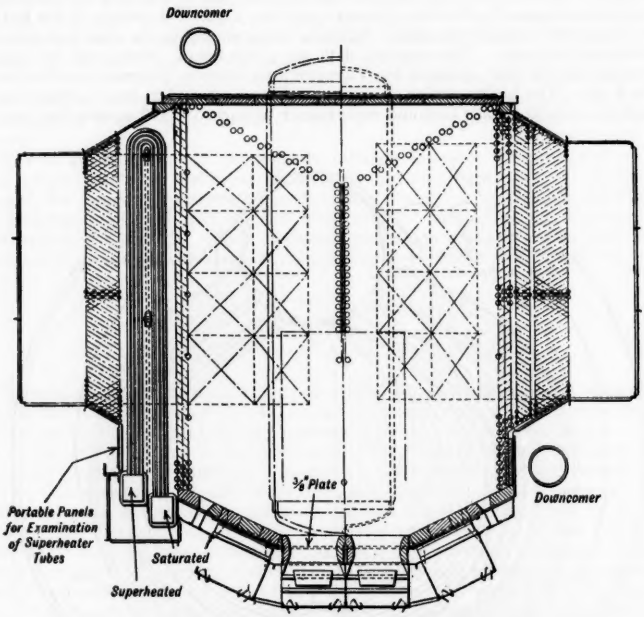


FIGURE 2.—SECTIONAL ARRANGEMENTS OF JOHNSON BOILER.

In Figure 3 we illustrate graphically the results obtained over a wide range of loads. The points of interest in these curves are the variations in superheat, and the relationship between the boiler efficiency and the rate of oil burning. In this connection it may be stated that the radiation and unaccounted-for losses were of the order of 3 to 4 per cent, so that there is every reason to anticipate that the efficiencies which would be registered in an actual, relatively air-tight stokehold would be somewhat higher than those recorded in the present tests.

At the high rates of forcing during the foregoing tests, the furnace temperature, which is always difficult to determine with accuracy, was somewhat less than occurs in furnaces having a large area of refractory, and there was a notable absence of priming, whilst the superheat temperature was steady throughout. At fractional loads, it was found possible, by adjustment of the dampers to obtain a considerable increase in the degree of superheat. It was demonstrated that no difficulty need be apprehended in the removal and replacement of fire tubes, it being possible to insert them through the furnace openings and place them in position, the flexibility of the tubes simplifying the operation.

Naval Boiler : Result of Tests.

	Designed load.	Overload.
Actual evaporation	166,320 lb./hour	192,122 lb./hour
Equivalent evaporation (from and at 212 deg. Fah.)	203,500 lb./hour	236,000 lb./hour
Steam pressure	300 lb. sq. in.	300 lb. sq. in.
Steam temperature	627 deg. Fah.	637 deg. Fah.
Feed temperature	183 deg. Fah.	183 deg. Fah.
Total fuel burned	14,069 lb./hour	17,140 lb./hour
CO_2	13·4 per cent.	13·44 per cent.
Funnel temperature	797 deg. Fah.	880 deg. Fah.
Air pressure at fronts	3·8in. W.G.	5·62in. W.G.
Lb. of fuel per burner	1,407 lb./hour	1,714 lb./hour
Ratios ;		
Evaporation per sq. ft. of generating surface :		
Actual	18·45 lb.	21·3 lb.
From and at 212 deg. Fah.	22·55 lb.	26·2 lb.
Evaporation per unit of boiler weight	1·3	1·5
Fuel burned per sq. ft. of generating surface ..	1·56 lb.	1·9 lb.
Fuel burned per sq. ft. of total heating surface ..	1·3 lb.	1·58 lb.
Fuel burned per cu. ft. of combustion chamber, volume	15·55 lb.	19·0 lb.

The weight of this boiler with hot water is slightly under 57 tons, so that an evaporation of 86 tons per hour corresponds to an output weight ratio of 1.5. With the experience gained as a result of these tests and various experiments made, the designers anticipate that it will be possible to make refinements in design which will secure an improvement in this ratio. It is interesting to note that the total weight of brickwork in this boiler does not exceed 3.5 tons.

A feature of the test arrangements was the very complete equipment for maintaining the boiler feed and oil fuel supply, and dealing in two large condensers with the large volume of steam generated.

The tests were conducted under conditions arranged to approximate as closely as possible to those obtaining at sea. The boiler was housed in a corrugated iron building in which the atmosphere was kept at 5.63 inches

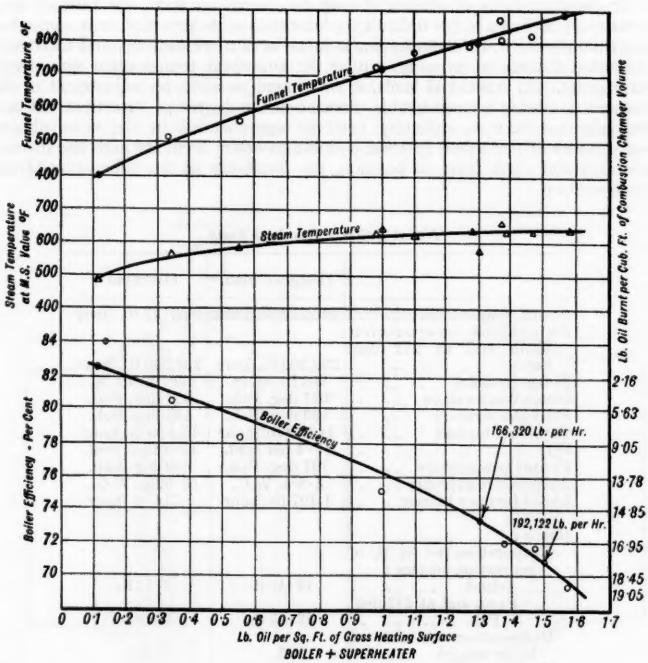

Funnel Temperature °F
Steam Temperature at H.S. Valve °F
Boiler Efficiency - Per Cent
Lb. Oil Burnt per Cub. Ft. of Combustion Chamber Volume
Funnel Temperature
Steam Temperature
Boiler Efficiency
168,320 Lb. per Hr.
192,122 Lb. per Hr.
Lb. Oil per Sq. Ft. of Gross Heating Surface
BOILER + SUPERHEATER

FIGURE 3.—TEST RESULTS.

W.G. above atmospheric pressure. The air for combustion was not pre-heated. The feed water was in continuous circulation; after being pumped into the boiler and evaporated it was discharged into the surface condensers and thence to the feed tanks and feed pump suction. The auxiliary plant was as follows: Two Weir air pumps; two condensers, provided by J. S. White and Co., Ltd.; four fans; two Weir fuel pumps; one feed heater. The steam was taken from the boiler to a de-superheater, which, in turn, exhausted into the two condensers. The two air pumps then discharged the condensate into calibrated tanks to which the feed pump suction was coupled.

The fuel oil and water were taken from special calibrated tanks which had been tested by the Admiralty. The tests were carried out using instruments approved by the Admiralty, which included a "Mono" gas analyzer, a Cambridge multiple temperature indicator, and Budenberg pressure recorder.—"The Engineer," Dec. 29, 1933.

FULL SCALE TRIALS ON A DESTROYER.

Performance of the U.S.S. "Hamilton" Compared with Model Tests at the Washington Tank.*

The fundamental endeavor of a model basin is to predict accurately the performance of ships and the effect of variables on this performance. Whether these aims are fully realized depends on the accuracy of self-propulsion model tests in indicating full-scale performance and on corresponding reproduction of the effects of variables from model to full scale. In an endeavor to arrive at more reliable and accurate results for high-speed vessels the U. S. experimental model basin developed larger self-propulsion dynamometers to drive the models at full speed. Since model propellers do not cavitate behind a self-propelled model the variable-pressure water tunnel was designed and built to investigate the effect of this variable on thrust, torque and revolutions.

In an effort to determine the extent to which these endeavors were being realized in its work, especially with regard to the design of high-speed vessels, extensive full-scale trials were undertaken on a destroyer. Because of its availability, the U.S.S. *Hamilton* was selected for this research. The principal dimensions of the vessel are as follows:

Length between perpendiculars, feet	310
Breadth, extreme, feet	31
Mean draught (on trial), feet and inches	9–6
Displacement, tons	1250
Block coefficient	0.472
Prismatic coefficient	0.620
Midship section coefficient	0.764

The propellers are solid, three bladed, of manganese bronze, with modified elliptical profiles and ogival sections. The faces of the blades are machined to a helicoidal surface throughout. Their particulars are as under:

Diameter, feet and inches	9–2
Pitch, feet and inches	10–2
Hub diameter ratio	0.168
Pitch ratio	1.109
Thickness ratio	0.054
Disc area, square feet	66
Projected area/disc area	0.55
Minimum clearance between hull of vessel and tip circle, inches	28⅜

In order to secure as full and as reliable data as possible, experiments were conducted on a self-propelled model of the vessel in the Washington Tank. Open water tests were carried out with propeller models to determine the thrust, torque and efficiency characteristics; and tests were made with model propellers in the variable pressure water tunnel in order to determine the cavitating regions.

Points observed prior to and during the ship trials included the maintenance of a constant displacement and the careful measurement of the actual screws. The work also included the installation of a Sperry Gyro pilot; a Sperry course recorder; a pressure speed log for measuring directly the speed

* Précis of paper read before the Society of Naval Architects and Marine Engineers, New York, by Commander H. E. Saunders and Lieutenant-Commander A. S. Pitrie.

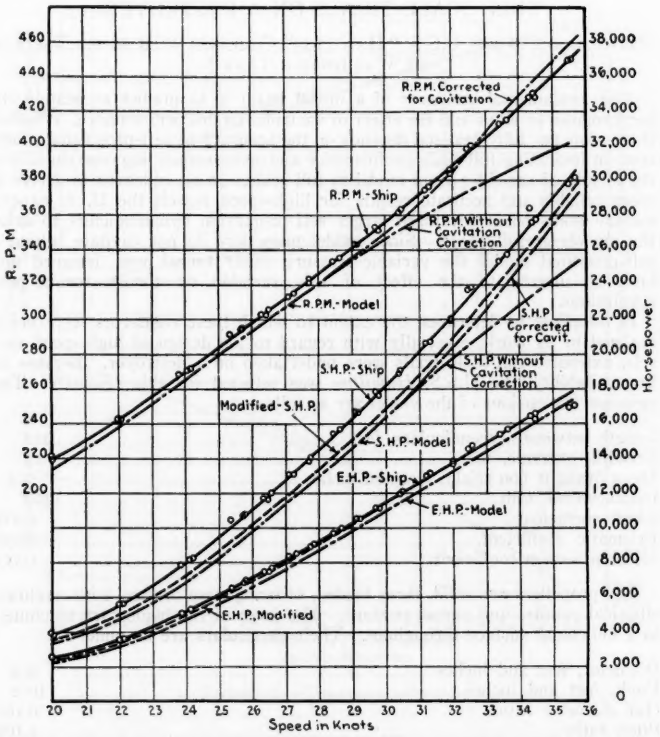


FIGURE 1.—U.S.S. "HAMILTON" STANDARDIZATION TRIALS, MAY 9, 1933 (DATA CORRECTED FOR WIND AND CURRENT AND TO A DISPLACEMENT OF 1250 TONS).

through the water; Ford torsionmeters; special thrustmeters; instruments for measuring trim; revolution recorders; anemometers for measuring wind velocities; swinging vane for measuring wind direction; and a mechanical device for indicating the actual rudder angle. The length of the approach to the "course" was increased from one to three miles.

Standardization trials were carried out in October, 1931, and also in May, 1933, and these trials gave practically identical results. The results of the 1933 trials are shown in Figure 1 above. The table gives a comparison between model prediction and trial performance taken from the 1933 results.

DISCUSSION OF RESULTS.

(1). *General.*—From a study of the table it is seen that for speeds up to and including 30 knots the excess of trial over model values of revolutions, effective horsepower, and shaft horsepower is decreasing. The increase of

COMPARATIVE PERFORMANCE, MODEL AND TRIAL RESULTS OF MAY, 1933.

The differences are expressed in percentage increase over predicted values.

Speed	Revolutions			Effective horse-powers			Shaft horse-powers		
	Model	Ship	Difference	Model	Ship	Difference	Model	Ship	Difference
20	214	220	2·8	2,200	2,500	13·6	3,400	3,700	14·7
21	225	231	2·7	2,650	2,950	11·3	4,000	4,600	15·0
22	238	242	1·9	3,200	3,600	12·5	4,800	5,600	16·7
23	251	255	1·4	3,900	4,300	12·8	5,800	6,700	15·5
24	265	270	1·9	4,680	5,160	10·3	6,900	8,000	15·9
25	279	284	1·8	5,600	6,100	8·9	8,300	9,500	14·5
26	293	298	1·7	6,700	7,100	6·0	9,850	11,100	12·7
27	307	312	1·6	7,700	8,150	5·9	11,400	12,700	11·4
28	320	326	1·7	8,750	9,200	5·2	13,000	14,500	11·5
29	335	340	1·5	9,900	10,300	4·0	14,600	16,300	11·6
30	350	355	1·4	11,050	11,450	3·6	16,350	18,200	11·3
31	366	370	0·9	12,200	12,650	3·7	18,300	20,200	10·4
32	385	387	0·5	13,350	13,800	3·4	20,400	22,200	8·8
33	408	404	− 0·9	14,500	14,950	3·1	22,600	24,350	7·8
34	429	421	− 1·9	15,600	16,100	3·2	25,000	26,600	6·4
35	448	440	− 1·8	16,700	17,300	4·2	27,400	28,900	5·5

trial shaft horsepower over model shaft horsepower is higher than the corresponding rate governed by the increase in revolutions. For example, at 22 knots the ship R.P.M. is 1.9 per cent higher than the predicted model value. For this increase in R.P.M. the ship shaft horsepower should be approximately 9 to 10 per cent higher than the model shaft horsepower. Actually it is 16.7 per cent higher. For speeds above 30 knots the same general characteristic is found as noted above. Here, however, there is a wider discrepancy in the increase in shaft horsepower from that determined by the rate of increase in revolutions. For example, at 33 knots the ship R.P.M. is 0.9 per cent lower than the model value. For this decrease in R.P.M. the ship shaft horsepower should be approximately 5 per cent less than the model value. Actually it is 7.8 per cent higher.

(2). *Effective horsepower.*—Since cavitation has no influence on the effective horsepower, Figure 2 has been prepared to analyze this factor. The thrust horsepower curve—*obtained from the measured thrust and the speed through the water*—when plotted is remarkably fair, with little scattering except for the two highest spots. If the data from the runs belonging to these two highest spots are free from appreciable error, the conclusion is that at complete breakdown of the propeller in cavitation the thrust falls off. This means that in relation to the resistance the thrust-deduction factor has been materially reduced.

By applying the thrust-deduction coefficient obtained from model experiments to the thrust horsepower a trial effective horsepower curve has been determined. Comparison of this later curve indicates that the frictional resistance is greater than has been anticipated. In the model calculations the frictional resistance was computed by Geber's method with a factor of 1.14, or a 14 per cent allowance for roughness. In view of the fact that the vessel was approximately 12 years old at the time of these trials, the under-water surface was undoubtedly more rough than when the vessel was new. This condition would normally result from corrosion and the continual application of bottom paints over such a long time. What improvement scaling would make in the character of the surface is problematical. Further, the fact that the vessel was out of dock a total of 11 days with five inactive days prior to the 1933 trials would increase the frictional resistance by about 1½ per cent. Because of this it would not be abnormal to expect a total increase of 10

per cent in the frictional resistance. Accordingly, using a factor of 1.25, the frictional resistance was recalculated (by Geber's method) and modified effective horsepower values obtained.

(3). *Shaft horsepower.*—Since cavitation is experienced in these trials, the shaft horsepower performance is treated under two classifications: that without and that within the region of cavitation. It is assumed that cavitation does not set in until 30 knots. Accordingly, the performance up to this point is analyzed first. Applying the propulsive coefficient for each speed as determined from model experiments to the modified effective-horsepower curve, there is obtained a modified shaft-horsepower curve, as given in Figure 1. A comparison between the modified shaft horsepower and the

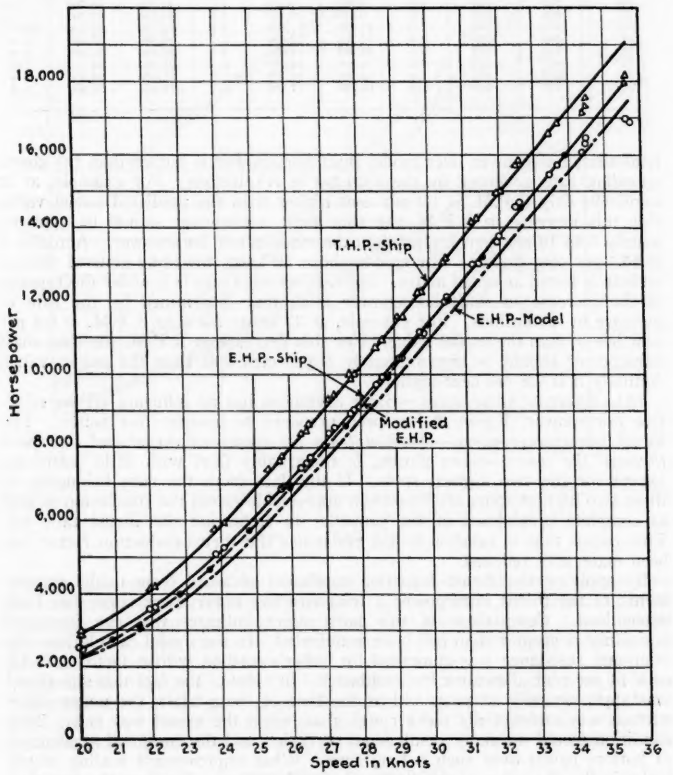


FIGURE 2.—ANALYSIS OF EFFECTIVE HORSEPOWER FOR STANDARDIZATION TRIALS OF THE "HAMILTON," MAY 9, 1933.

shaft horsepower measured on trial shows that the trial powers are approximately 6.5 per cent higher than the modified powers.

(4). *Propeller performance.*—To reconcile this difference it is essential to analyze the performance of the two propellers. Since both torque and thrust were measured, it is possible to calculate the efficiency of each propeller. On doing so it is found that the ship propellers are approximately 7 per cent less efficient than the model screws. This difference is attributed to the rough surfaces of the backs of the blades.

(5). *Speed through water.*—A comparison of the speeds as derived from the trial analysis and those measured by the pressure speed log indicates that for the 1933 trials the speeds measured by the log are 1.72 per cent low, that is, in comparison with the speeds obtained from observing the time of transit and corrected for current effect but not wind effect. Just what percentage of error may be present in these latter speeds is unknown. Hence the comparisons established are purely relative and not absolute. For this reason the speeds obtained from the pressure log can be considered within the usual engineering limits of accuracy.

(6). *Current on trial course.*—It was found that the spots recorded for the 1933 trials showed a southerly river current averaging from 0.10 to 0.30 knots.

(7). *Effect of rudder.*—During the 1933 trials the vessel consistently carried right rudder. Considering that the starboard propeller produced a greater thrust than the port propeller, this condition might have been anticipated. To what degree this condition is reflected in the increased resistance is problematical. The induced drag is appreciable and might account for some of the increased resistance observed from the trial data.

(8). *Variations in observed data.*—Observations were taken every 15 seconds and the variation in the revolutions observed. The fluctuations in the revolutions per minute and power do not have an influence of the same order on speed. This means that because of the ship's momentum its speed cannot follow rapidly the variations in revolutions and power.

(9). *Cavitation.*—Figure 1 shows the usual corrections applied to a standard self-propulsion test for cavitation. The following table gives the percentage increase in revolutions and shaft horsepower for these corrections as determined from tests of the model propellers in the variable-pressure water tunnel at the experimental model basin:

Speed.	Revolutions.	Shaft horsepower.
28	0	0
29	0.3	0
30	1.2	0
31	2.2	1.7
32	4.3	3.6
33	7.7	6.1
34	10.3	9.7
35	12.8	12.4

The characteristic curves for the model propellers giving data in the cavitating region are shown in Figure 3. There is a close harmony between the actual results in this connection and those derived from these tests in the variable-pressure water tunnel.

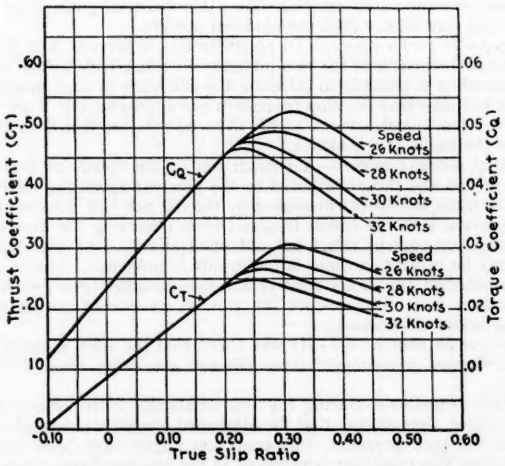


FIGURE 3.—CHARACTERISTIC CURVES FOR MODEL PROPELLERS NOS. 1214 AND 1215 (DATA FOR CAVITATING REGION DERIVED FROM TESTS IN VARIABLE-PRESSURE WATER TUNNEL).

Conclusions.—Full-scale tests when sufficiently extensive and carefully conducted will furnish data very nearly as accurate as can be obtained with model tests under laboratory conditions, provided corrections for all tangible sources of error are applied. These trials emphasize the necessity for measuring thrust.—"Shipbuilding and Shipping Record," Decembber 14, 1933.

VISIBILITY AT SEA.

As an outcome of extensive tests made on behalf of the Hamburg Institute for Physical and Biological Light Research, Prof. F. Dannmeyer makes a strong plea in "Hansa" for the general use of neodymium glasses or sight screens by navigators. The glass itself is of a blue-grey tinge and its characteristic property is to absorb yellow rays from the spectrum, thus giving greatly increased definition to the red and green. Trials conducted on the Elbe and in the North Sea and Baltic under summer and autumn conditions show that the neodymium glass, whether used in the form of spectacles or as a 12 inch by 6 inch window on the bridge, results in substantially improved visibility. In clear weather, the markings of a vessel are distinguishable at increased distances, and, in haze, colors can be distinguished when the vessel appears only as a grey mass to the naked eye. Even more important, the estimation of distance and course is greatly facilitated. Other facts observed are that vessels and navigation marks can be seen more easily and clearly against the rising or setting sun, owing to the elimination of the dazzling yellow rays; an extra mile or so of coast line becomes visible in estuaries in

hazy weather; and the visibility of green lights at night is improved. The problem of visibility in fog, however, has still to be solved, probably by indirect means, such as the conversion of an image initially formed by invisible rays.—"Shipbuilding and Shipping Record," December 14, 1933.

THE PROGRESS IN THE DEVELOPMENT OF NAVAL GUNS.

Since the war considerable progress has been made in the development of naval guns both as regards the matériel itself and its application.

One of the most interesting developments is the employment of the process known as "radial expansion." Guns were formerly made with an internal tube on which hoops were shrunk. Their construction was a long and difficult process and they were composed of a large number of parts. The 138-millimeter 1910 model had 12 parts and the 340-millimeter 1912 model 29 parts. Radial expansion consists in taking a tube, the ends of which are closed, and applying to it an internal pressure exceeding the elastic limit. The tube transversely assumes a permanent deformation, and when the pressure has ceased, the external layers form hoops on the internal layers. The strength of the gun is thereby increased. To perfect this process, which requires pressures of from 5 to 6000 kilograms with tight packing at the end plugs, was difficult and involved trials and errors. But the process is now established in practice and the resulting advantages are indicated by the following figures: The 155-millimeter 1920 model weighs 9.5 tons; the 155-millimeter Schneider monobloc radially expanded, with equivalent ballistic characteristics, weighs 6.5 tons.

There is also the relining. With the present high initial velocities guns are quickly worn out and, in consequence, after 100 to 150 rounds lose their accuracy. Instead of replacing the worn out guns, only their lining tube need be replaced. This is retubing. But the operation is a long and costly one under present conditions, with the liner driven in by force and rifled only after installation. Since the war relining (chemisage) has been invented. This consists in placing the thin liner in the gun with a slight clearance. In firing, it is elastically deformed and grips the gun, which temporarily increases the strength, after which it resumes its original position. This tube or liner, since it does not grip the gun, can be readily removed and replaced. It only takes an hour in the case of calibers up to 138 millimeters, three hours in the case of calibers of 138 to 203 millimeters, and less than a week in the case of calibers of more than 203 millimeters, while retubing took several months in the case of large calibers. High-grade linings, radially expanded, cost about one-third of the price of a new gun—300,000 francs as compared to one million for a 340 caliber.

Thus the weight of the guns is reduced, their construction is simpler, and they can be quickly and economically repaired when they are worn out. Equally important is the progress which has been made in increasing the rate of fire, due to the reduction of the amount of time lost at various stages in operating the breech and in loading.

The naval breech mechanism of the standard type is a screw breech which is opened in three movements: unlocking, withdrawal, swinging clear. The translational movement has been eliminated by the employment of a sliding block type of breech mechanism. Furthermore, in order to increase the rapidity of the operation, it has been made automatic. The force of the recoil at the return to battery opens the breech and compresses a spring. In loading, the moving cartridge case releases this spring, the force of which is used for closing the breech. The quick-firing anti-aircraft guns, firing with

cartridge cases, even have breech mechanisms which are opened by a single movement; and in some cases the closing of the breech effects the discharge.

Guns of small caliber, up to 100 millimeters inclusive, have ammunition in cartridges which are loaded by hand. In the case of larger calibers considerations of weight and space make it necessary to separate the projectile from the charge or from the cartridge. The projectile is now pushed into place by an automatic rammer, which is energized on recoil and the arrangement of the loading trays also tends to reduce the dead time.

In the case of small and medium calibers, the guns can be readily loaded at any angle of elevation. In the case of large guns, it becomes very difficult to ram the projectile in position and seat it, regardless of the angle of elevation, for it is necessary to employ very powerful springs, which will act with too great a force when the gun is loaded in the horizontal position.

The attempt to gain rapidity by automatic means is not without inconvenience and even danger. In a turret, the pointers stationed forward of the gun do not see what is going on near the breech mechanisms. The gun elevating and training motors and ammunition hoist motors, the air and water blow offs (gas ejectors), the loading, and the sounding of firing orders make a deafening noise. It is necessary to take into account the resulting nervousness, a nervousness which may lead to operations which are not properly timed, such as premature firing, opening of the breech before firing, etc. In order to avoid this, it was necessary to install mechanical safety devices. But new complications were produced in a matériel which is already terribly complicated. In trying to eliminate accidents we have to depend on numerous complicated mechanisms which frequently do not function properly.

The same inconvenience is encountered in fire direction and control, when remote fire control is employed and the angles of elevation and train are transmitted to the gunners, who lay the guns according to the instructions received, or in the case of remote control when the remote control station, by means of servo-motors, controls the guns from a distance. In no case—except in case of damage to the remote control system—is local telescopic pointing used. It is the angle of train and the angle of elevation which serve as coördinates and which the remote control station transmits to the guns. An officer stationed aloft on the vessel is thus the sole judge of a correct point of aim on the target and electrically fires all the guns. Excepting when the operating mechanism does not work, the difficult work of control pointing is performed by one man, who, naturally, must be carefully selected, highly trained, and very accurate. In short, nearly the entire fighting efficiency of the vessel depends on him. It also depends on the reliable functioning of the special electric fire control appliances, the complexity of which during the last few years has increased to an almost incredible degree. To quote only one example, at least four different currents are used for the motors, the various transmission gears and the particular lighting. This requires converter sets with all their fittings. Some slight damage to these, which is always possible, may make all the guns ineffective. This is the price of our wonderful inventive ingenuity. And besides extremely careful construction, these technical developments require operating personnel, in every grade, with a more and more advanced technical training and unfailing reliability.—"Journal de la Marine Le Yacht," 30 December, 1933.

ASSOCIATION NOTES.

ELECTION OF OFFICERS FOR 1934.

The following have been elected officers of the Society for the year 1934:

President:

Admiral William H. Standley, U. S. Navy.

Secretary-Treasurer:

Commander H. B. Hird, U. S. Navy.

Council:

Captain Henry Williams (CC), U. S. Navy.
Captain H. R. Greenlee, U. S. Navy.
Captain C. A. Jones, U. S. Navy.
Commander J. J. Broshek, U. S. Navy.
Commander H. F. Johnson, U. S. Coast Guard.
Mr. A. F. E. Horn.
Mr. J. E. Burkhardt.

MEMBERSHIP.

The following have joined the Society since the publication of the last previous JOURNAL:

NAVAL.

Colton, E. B., Lieutenant Commander, U. S. N.
Deming, Raymond A., Lieutenant Commander, U. S. N.
Grove, George W., 200 Madison Ave., New York, N. Y.
Richey, Thomas B., Commander (CC), U. S. N.
Schetky, Gerald L., Lieutenant Commander, U. S. N.
Shute, Corben C., Lieutenant, U. S. N.

CIVIL.

Bull, R. A., 541 Diversey Parkway, Chicago, Ill.

ASSOCIATE.

Cooper, Charles G., 819 Mills Building, Washington, D. C.
Forde, Ivan S., 30 Church St., New York, N. Y.
Reichel, W. A., 754 Lexington Ave., Brooklyn, N. Y.